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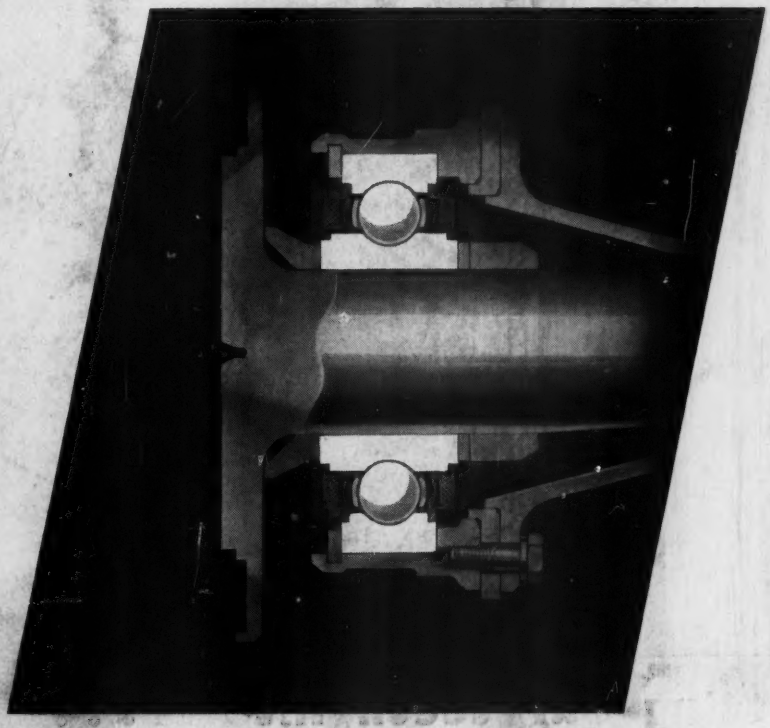
AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 40 No. 523 JANUARY, 1950 PRICE: 2s. 6d.



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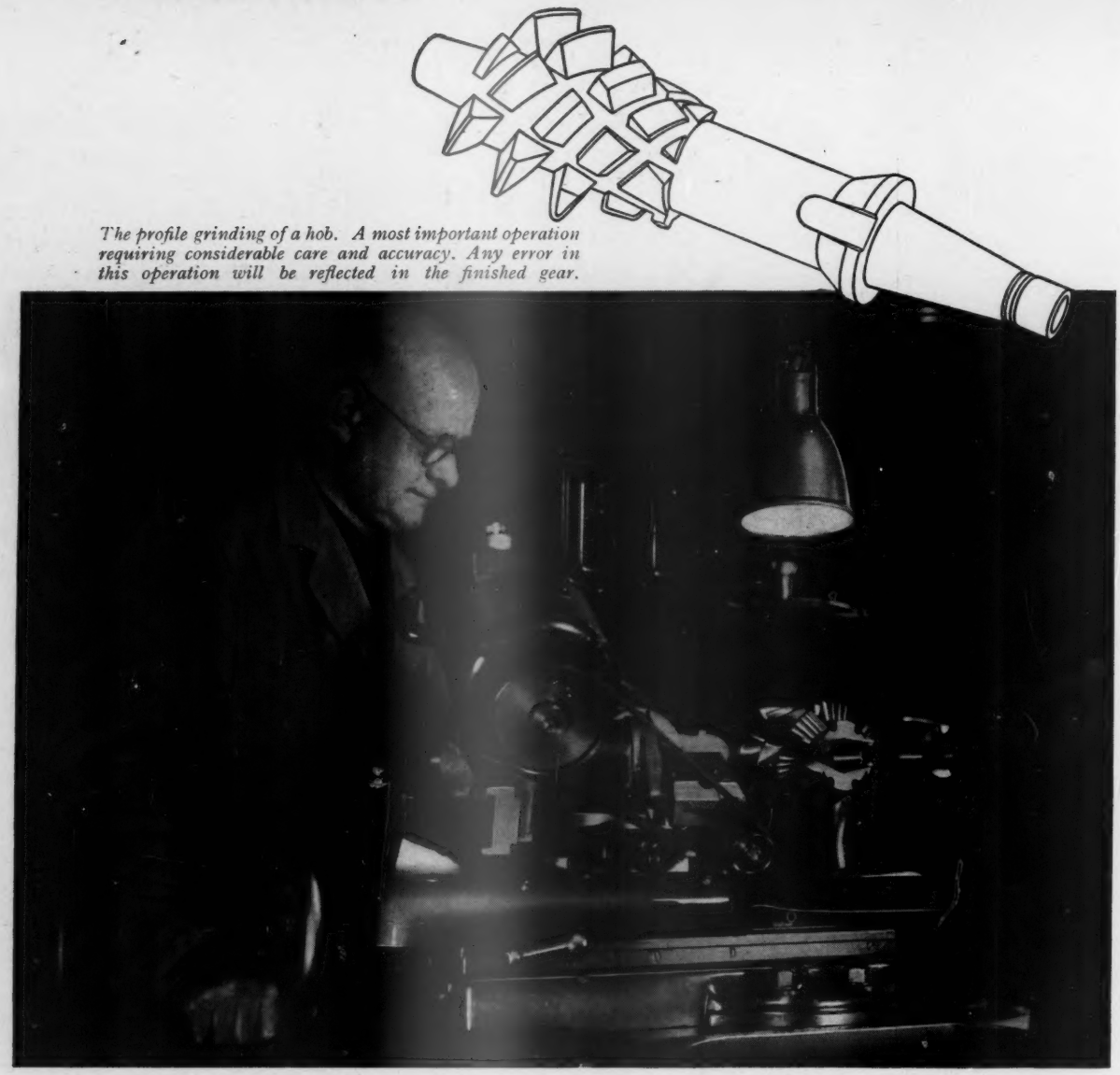

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AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

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Publishers: ILIFFE & SONS LTD., DORSET HOUSE, STAMFORD STREET, LONDON, S.E.1
Telegrams: Sliderule, Sedist, London Telephone: Waterloo 3333 (60 lines)

COVENTRY:	BIRMINGHAM, 2:	MANCHESTER, 3:	GLASGOW, C.2:
8-10, CORPORATION ST.	KING EDWARD HOUSE, NEW ST	260, DEANSGATE	26B, RENFIELD ST.
Telegrams: Autocar, Coventry	*Telegrams*: Autopress, Birmingham	*Telegrams*: Iliffe, Manchester	*Telegrams*: Iliffe, Glasgow
Telephone: Coventry 5210	*Telephone*: Midland 7191-7	*Telephone*: Blackfriars 4412	*Telephone*: Central 4857

PUBLISHED MONTHLY—SECOND THURSDAY

Annual Subscription: Home and Overseas £1 17s. 6d., including the Special Number; U.S.A. and Canada $6.00

VOL. XL. No. 523 JANUARY, 1950 PRICE 2S. 6D.

Productivity Measurement

ALTHOUGH some means of measuring productivity is essential in order that the comparative efficiencies of different departments or processes may be computed, this subject is one that has hitherto received little study. In January, 1949, however, the Institution of Production Engineers and the Institute of Costs and Works Accountants formed a joint committee with two objects, (a) to devise the best means of measuring comparative productivity and (b) of applying these methods for the purpose of increasing productivity. This committee has now issued an interim report.

At the outset, the committee found that there was no previous work to which they might refer. Certain studies had been made by economists, but their findings were of far too general a nature to be of use either to managements or to the supervisory staffs in the shops. It was also found that although many firms have their own methods of measuring productivity, they invariably referred to a finished product. They could not, therefore, be used for comparing the efficiencies of departments that dealt with only part of the manufacturing process. Much less could they be used for inter-firm or inter-industry comparisons.

Shortcomings

In the course of the researches, some disturbing facts were brought to light. They included:—

(a) A lack of knowledge of machine utilization in British industry, and it became evident that the standard of machine utilization is low. If it were improved, a significant contribution would be made to increased productivity.

(b) A lack of cost and statistical information available at supervisory level. Very often through poor presentation and lack of understanding, much of the existing information is ineffective.

(c) A serious lack of agreed principle in the application of time study methods. Present time study standards suffer in most cases from serious deficiencies for the purpose of comparison between departments and firms. If uniformity were introduced into the methods of arriving at such standards, valuable data would be available.

Early in the investigation the committee decided that comparative assessment of productivity on the basis of product is impossible. Processes, however, are common not only to different departments of a single organization but also to different firms and different industries. It is therefore obvious that by using a process rather than a product basis, a much greater range of comparison becomes possible. The greater the range of comparison, the more useful the standard of measurement.

By the relation of the many processes necessary for the production of a particular product to a common standard, it may well be possible to arrive at some definite conclusion regarding the productivity of a firm as a whole, and eventually of an industry as a whole. Further, as the committee stress, emphasis on process rather than product is more in line with the thinking of the departmental head or foreman. Indeed, one great advantage of process measurement is that standards of comparisons can be set for use not only by top management, but by all grades of supervisors, foremen and individual workers.

The Process Basis

To determine the value of using process as a basis, the co-operation of four important firms in the London area was obtained. Each of these firms had its own production engineering and cost accountants departments, and the management techniques are well in advance of standard practice. A particular process, the operation of a group of automatic lathes, was chosen for study.

Probably the most significant fact brought to light by the investigation was the low degree of plant utilization. In one case it was as low as 69 per cent. Since the four organizations concerned in the investigation are above the average for efficiency it can only be concluded that throughout British industry the degree of plant utilization is much lower than it should be. The report in fact states that a study, limited to machine utilization throughout British industry, should provide valuable information. If machine utilization of the more inefficient firms were brought up to the level of the best, there would be a very

considerable increase in all round productivity at very little extra cost.

Although this is only an interim report, it is nevertheless a valuable contribution to a hitherto neglected subject. It should be studied by all managements, production engineers and supervisory grades, and all who do so will await further reports with interest.

Production Methods

ALTHOUGH a certain amount of good will no doubt result from the visits that the various Productivity Teams have paid to the United States, there are also dangers which should not be overlooked. Merely to introduce American practices into this country will be far from sufficient. That there is much to be learned from American production methods and practices is true, but at all costs we must avoid becoming mere imitators. There is some danger of this, as may be seen from the design trend of certain British motor car bodies. In all too many cases, distinctively British design has given place to copies of American fashion, without any regard to the size and type of vehicle, or the general suitability.

Advantage must obviously be taken of American or any other developments. It is scarcely possible, however, to overstress the fact that Britain can continue as a leading industrial nation only if there is initiation as well as imitation. Imitation may be the sincerest form of flattery, but it is also the surest way to industrial stagnation. Those who merely imitate must lag behind those who initiate.

Rival Productivity

Much has been said and written on both sides of the Atlantic about the wide and unfavourable differences between British and American productivity, and unfortunately most of this is true. It must not, however, be forgotten that even in quantity production we are not necessarily dependent upon American brains for success. Given the requisite encouragement and support, the production engineers of this country need not fear comparison with those of any other country. In this connection, two developments within the past eighteen months should not be overlooked. Each concerns the manufacture in this country of products originally developed in America.

In one case, apart from two special purpose machines, the machines employed and the production layout were conceived in this country. American engineers from the plant where the product was developed, freely admitted that the British layout was at least 30 per cent more efficient than their own. In the second case, in a period of months, the British engineers had devised a production method that had always appeared desirable to the Americans, who, however, had not been able to make it practicable over a period of years. These are not completely isolated examples.

It is essential that British engineers of all classes should avoid complacency, and hard economic facts will probably prevent any such development. At the same time it is important that every precaution be taken to avoid a feeling of inferiority. Nothing is more likely to foster such feelings than an uncritical acceptance of practices and methods, merely because they are American. After all, American dominance in the Western world has been created by the recent war, coupled with a mistaken British post-war policy. It is not necessarily more permanent than any other situation, and this temporary dominance must not be allowed to cloud the vision unduly.

Quality

DISREGARDING the unstable economic condition of the world at present, there are, nevertheless, two points that will be generally conceded. They are, the vital importance of quality and also of cost. Quality, despite arguments to the contrary, should be wholly within control. Cost, on the other hand is only partly within the control of the industry. These two factors will, however, play an important, if not decisive, part in the prosperity of the industry.

British Standards

In general, so far as designs and specifications are concerned, British automobiles are of good quality. Unfortunately, in too many cases the final product is not up to the requisite standards of finish and reliability. This is not due to any deliberate sacrifice of quality in order to obtain increased output. So far as can be gathered it arises from two sources. The first is that workers generally now have little pride in their work, while the second arises through failure in final inspection. There is a quite general falling off in hitherto accepted standards.

The second cause is probably the more important since the necessary correction should be more easily applied and should largely correct the first. In this connection it is worth while noting the experience of one organization. Immediately after the war the management was seriously perturbed by the lower standard of work that was being produced. To offset this, they increased the inspection staff by more than 50 per cent, the increased staff operating mainly in the sections suspected as being chiefly responsible for the falling off in quality.

The management naturally gave the fullest possible support to the Inspection Department, and it has been possible gradually to reduce the ratio of inspectors to operators to the pre-war figure and yet retain quality.

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THE STRØMMEN BUS CHASSIS

A Norwegian Layout Embodying Four-wheel Independent Suspension

THIS interesting vehicle is assembled by A/S Strømmens Vaerksted, Strømmen, Norway, and is designed for operation in that country. The components, largely British, include the flat underfloor Leyland diesel engine, described in *The Automobile Engineer* for September, 1949, and independent suspension all round is embodied.

The chassis was planned for use as a single-deck vehicle, 32ft 2in overall and 8ft wide, with a 16ft 8¾in wheelbase. It is of integral or "chassisless" construction with independent suspension of all four wheels. Laminated springs are used and the installation largely eliminates lubrication points, nor are the springs torsionally stressed.

The power unit is a Leyland 125 b.h.p. horizontal six-cylinder in-line direct injection oil engine of 600 cu. in, embodying a hydraulic coupling and driving through a power-actuated epicyclic four-speed gear box to a differential unit suspended from the underframe. The Lockheed hydraulic brakes have air-pressure servo. Lightness of steering is stated to be a feature, which has been attained by centre-point layout, in combination with deeply dished front wheels which enshroud the swivel pin assembly. Large section single tyres are fitted on the rear wheels.

SPECIFICATION

ENGINE : Leyland direct injection diesel with toroidal cavity pistons. 120 x 140 mm bore and stroke, 9.183 c.c. (600 cu in). Six cylinders in line, horizontal. 125 b.h.p. at 1,800 r.p.m.

CLUTCH : Vulcan type fluid coupling.

GEAR BOX : Wilson epicyclic, preselective change with air-pressure servo hydraulic control. Four forward speeds and reverse. Forward ratios, 1, 1.56, 2.36 and 4.15 to 1.

TRANSMISSION : Open cardan shaft with Layrub joints.

DIFFERENTIAL : David Brown frame-mounted under-worm drive and differential unit. Ratio 5.8 to 1.

REAR SUSPENSION : Live stub axles driven from differential unit by Mechanics cardan shafts.

FRONT SUSPENSION : Fabricated tubular wish-bones.

SUSPENSION : All wheels independently sprung. Cantilever rear springs. Transverse front spring.

BRAKES : Lockheed. Hydraulic operation with air pressure servo. 17in diam drums on all wheels. Front shoes 4in wide. Rear, 5in. Parking brake on transmission.

TYRES : Front, 10.00 x 20in. Rear, 12.00 x 20in on detachable steel disc wheels (single rears).

CONSTRUCTION : Integral "chassisless" welded steel underframe with riveted alloy floor and body.

Fabricated steel sections form the underframe, which is of welded construction throughout and incorporates all the anchorage points for the suspension system, the power unit and transmission components. The stub axles are carried at the outer ends of two wishbone structures fabricated from steel tubes and pivoted from longitudinal members of the underframe. The pivot bearings are widely spaced giving considerable fore and aft rigidity to the wishbones. Very little caster inclination is employed at the swivel pins, about one degree being found to provide a self-centring action to the steering.

Suspension is by a laminated spring mounted transversely under the framing and connected to the stub axle units by T-headed bolts in tension. The bolts pass upwards through holes in the ends of the spring master leaf so that the T heads which are case hardened, bear against its underside. This knife-edge bearing does not necessitate lubrication. The other end of the bolt passes through a lug incorporated in the wishbone, contact being through a radiused washer positioned

The Strømmen bus.

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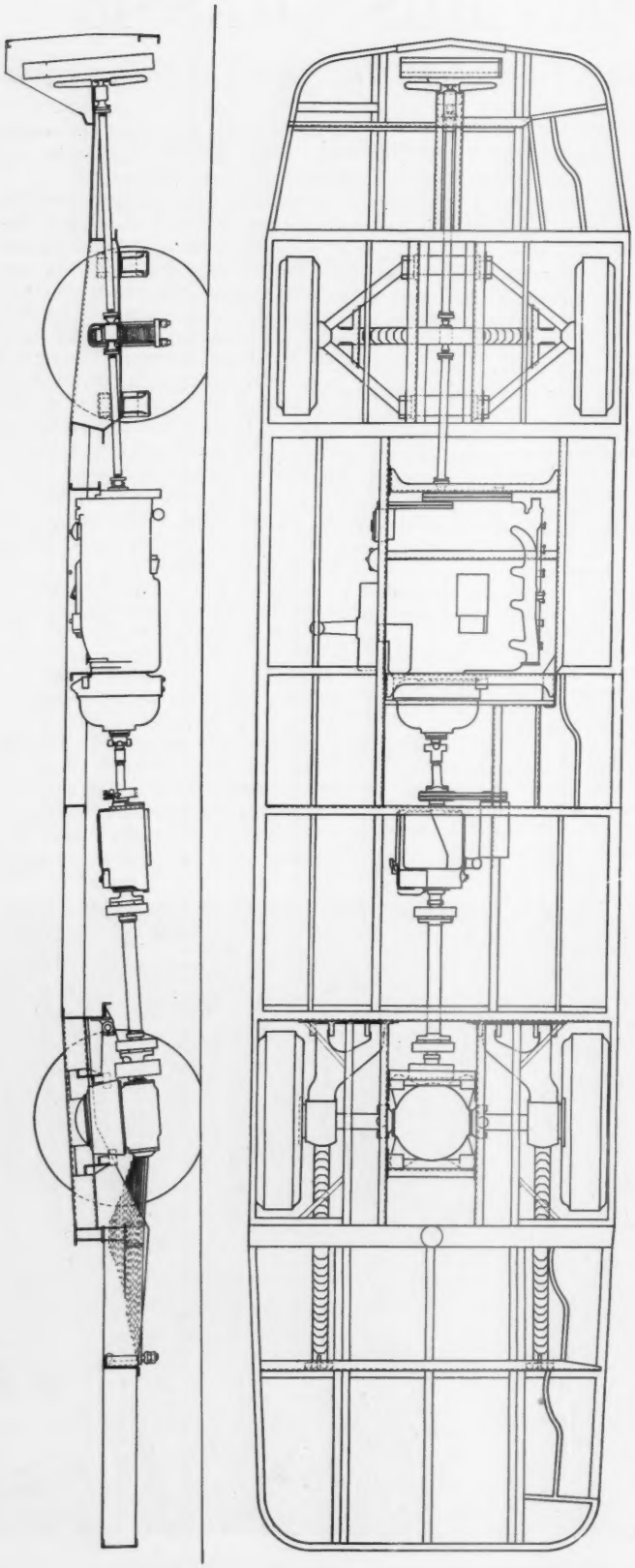


Layout of under-frame and mechanical components of the Strømmen bus.

by a nut and lock-nut. By adjusting these nuts the height of the body in relation to the wheels can be adjusted at front and rear independently.

At the rear, the live stub axle of each back wheel is carried in taper roller bearings in a cylindrical housing which forms part of a lever or radius arm pivoted to a transverse member of the underframe. The pivot bearings are widely spaced and thereby the arm is constrained to move in an arc in a vertical plane parallel to the vehicle centre line. Movement is controlled by a semi-elliptic spring mounted as a cantilever. That is to say, the inverted spring is trunnioned at its centre to the underframe, one end bearing on the axle and the other end being shackled or otherwise articulated to the frame.

In the Strømmen arrangement of rear suspension the application of the cantilever spring is unusual in that the entire spring is to the rear of the axle. Just why this lay-out was adopted is not altogether clear. It is possible the objective was to obtain fixing points distributed over a considerable area of the underframe and in particular beneath that part which greatly overhangs the rear axle. Whether this scheme offers any real advantage in this respect as compared with a conventional semi-elliptic arrangement seems debatable. It gives the impression that the distribution of support and the reduction of overhang is more apparent than real.

On the other hand, the lateral rigidity of the hub-carrying radius arm completely relieves the spring from driving and braking torque and eliminates all torsional requirements. This being so the spring serves only for suspension, and it can be designed accordingly. Further, since neither the spring itself nor its connections have to deal with torque, the forward end has been arranged to rest on a roller that is carried between lugs in the lower part of the hub housing. The rear end is articulated to the underframe by a height-adjustable knife-edge bearing T-headed bolt similar to that employed at the front suspension wishbones.

The roller on which the forward end of the spring rests, is mounted on needle roller bearings on a fixed pin. As load increases, the radius arm swings upwards, or the body sinks, thus shortening the effective length of the spring forward of the trunnion. This is due to the fact that the radius arm is pivoted forward of the axle while the cantilever spring is trunnioned to the rear. Though not immediately obvious, this is nevertheless one of the most important benefits of the unconventional rear suspension

Installation of Leyland engine and Sinclair fluid coupling.

scheme. By this effective shortening, the stiffness of the spring is increased with increase of load. When the vehicle is operated constantly at from half to full load, a single roller gives a satisfactory amount of variation. Where wider range is required, two rollers are incorporated in the axle housing. The first is in contact with the spring at light loadings while the second comes into action nearer the full load conditions, shortening the forward end of the spring by about two inches.

All the hubs are mounted on taper roller bearings. Those at the rear are mounted externally on the housings that enclose the live axles. The drive is transmitted through an upset flange that is clamped to the hub by studs and nuts. The pressed steel wheels are deeply dished so that the centre of the tyre tread is well over the bearing centre line in the case of the rear wheels, while in front the space available within the wheel permits the swivel pin to be in the vertical plane, so that the centre line passes approximately through the centre of the tyre tread, there being one degree of caster.

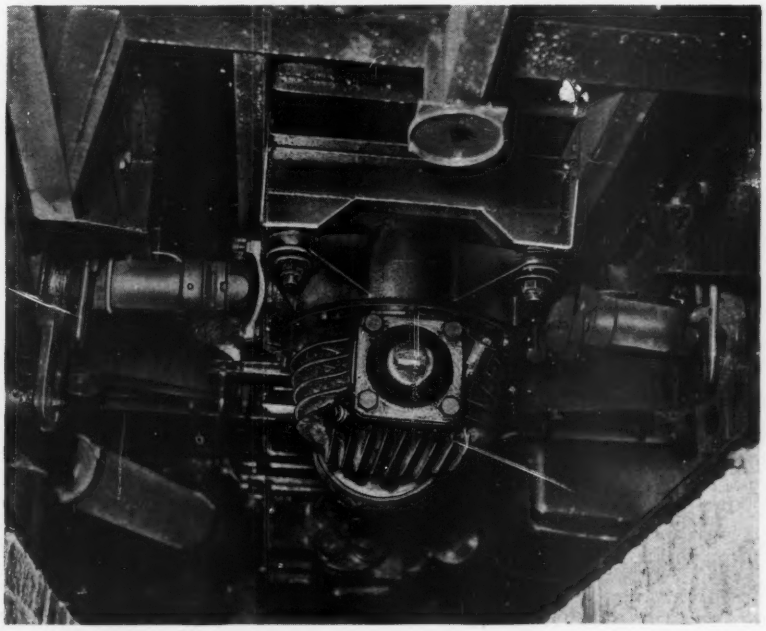
Ross cam and roller steering gear is fitted and the drag link is replaced by a parallelogram device. The cranking of the track rod is somewhat exaggerated in comparison with a conventional axle arrangement and this may account for a certain heaviness in the steering at very low speeds, as when manœuvring. Under normal running conditions the steering is sufficiently light.

The power unit and transmission are drawn from many sources, with British power equipment predominating. The engine is the flat or underfloor version of the Leyland 600 cu. in direct injection diesel which was exhibited at the last Earl's Court Show and was fully described in *The Automobile Engineer*, for Sept., 1949. It is suspended by the Leyland system of rubber bushed shackles, and is mounted at about the mid point of the wheelbase. The crankshaft is approximately on the fore and aft centre line of the vehicle and the cylinders are towards the right-hand side. Running forward to the front of the vehicle, where the radiator is situated, is a Layrub-jointed open shaft driving the fan.

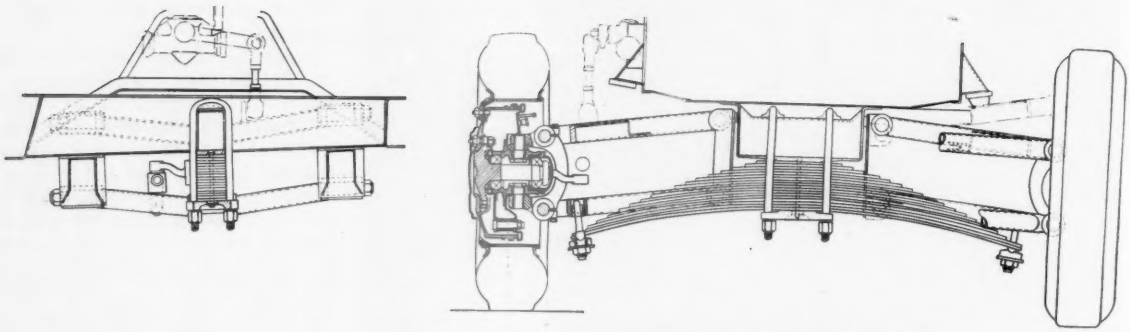
Integral with the flywheel is a Vulcan-Sinclair fluid coupling which is connected by a short clutch shaft to a Wilson pre-selector four-speed epicyclic gear box giving forward ratios of 1, 1.56, 2.36 and 4.15 to 1. Actuation of the gear brake-band bus bar is effected hydraulically with air-pressure boost. From the gear box the drive is by an open tubular Layrub cardan shaft to a David Brown differential unit which is suspended from the underframe by rubber insulated bolts. The drive is an underworm of 5.8 to 1 ratio.

Short shafts with Mechanics universal joints, one of which incorporates a splined slidable flange, couple the differential gears to the live axles. A contracting band brake is mounted on the nose piece of the worm shaft housing. The service brakes on the hubs are of the internal cylinder type Lockheed with 17in drums, the shoe width being 4in at the front and 5in at the rear. The hydraulic actuation is air-pressure boosted.

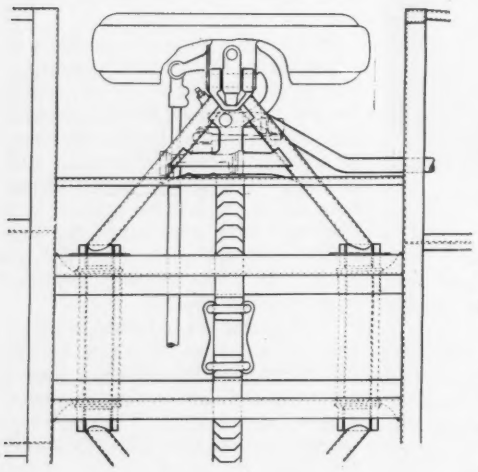
This system was adopted in preference to air operation alone because under extreme cold conditions there is always risk of the freezing of condensate in long air lines. The rear single wheels have 12·00 × 20in tyres while the fronts are 10·00 × 20in. Single rear tyres permit of the greatest possible width over springs, and in front, the transverse spring in line with the stub axles provides the maximum centre distance between the points of



Installation of David Brown differential unit showing cardan shafts.



Layout of independent front suspension.



articulation. This construction also assists maximum unrestricted steering lock, the bus with its 16ft 8¾in wheelbase having a turning circle of only 82ft, measured at the wheel tracks.

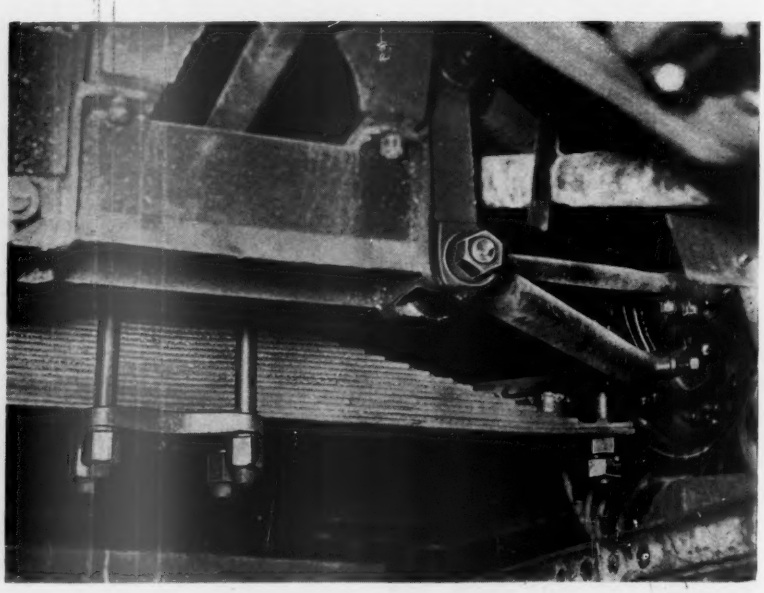
Body structure

To the steel underframe of the vehicle an aluminium floor is riveted, to which in turn the pillars and body sides are riveted, so forming a box girder. To this the mechanical components are assembled at a later stage. Thus the body structure is completed first after which the major components are added. This has been found to be the most effective procedure from the point of view of production. The mechanical components can, however, be fitted to the simple underframe and floor unit and at this stage constitute a completely self-contained chassis which can be driven as such. It is not a load carrier, of course, since the body itself is of the greatest importance in providing the necessary strength as a girder both in bending and torsion.

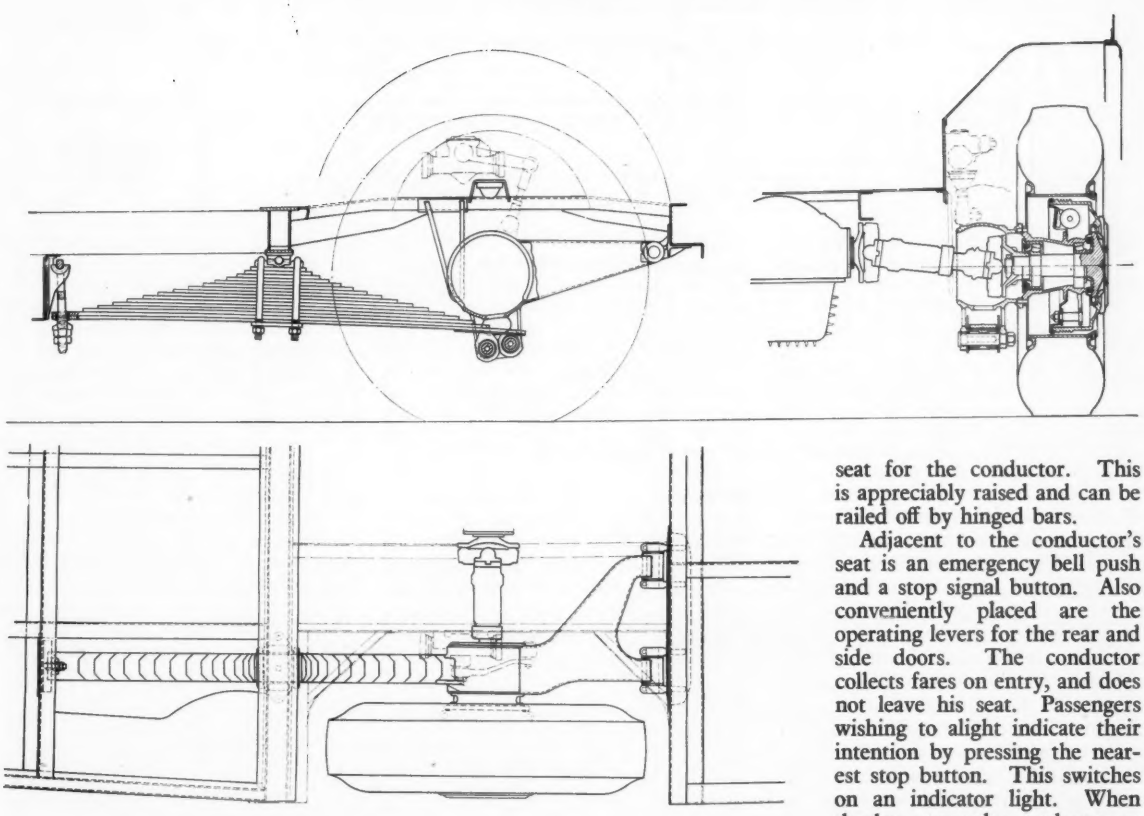
The single-deck body is an all-metal structure planned for economy of material and the avoidance of any fittings that cannot be regarded as functional. British practice is determined by the legal requirements as to seating, plus the allowance of a small number of standing passengers on sufferance. Norwegian conditions permit of a large number standing, with seats for a much smaller number. Accordingly the Strømmen bus is planned for 75 passengers, 51 standing and 24 seated. The gross load, therefore, is 12 tons 19 cwt, unladen weight 7 tons 5 cwt.

Three doors are provided on the right side, or more accurately four doors, in that the rearmost opening, which is the entrance, has double doors. This arrangement gives separate access to the rear standing space and to the main part of the interior forward of the rear wheels. A central door about midway along the side of the vehicle is for exit only. Forward of the front wheels is the remaining door also used as a passenger exit and for access to the driving compartment. All the doors are of the folding "knife-board" type and are operated by air pressure. The forward door is under the driver's control while the rear and central doors are controlled from the conductor's position.

Light alloy is used throughout the body structure, the pillars being of channel section with inwardly lipped flanges, which results in what may be described as an open-faced box section. This section gives full access to the panel riveting and provides a convenient system of conduits for the wiring which is very fully utilized since on each pillar is a push switch by means of which passengers signal to the driver to stop. The open face of the pillar is filled by an H section rubber strip sprung on to the inturned edges of the flanges. The windows are very deep and in combination with the narrow pillars give a very light interior and an avoidance of weight effect in the external appearance. The cant-



Front spring and wishbone assembly.



Arrangement of rear suspension and final drive.

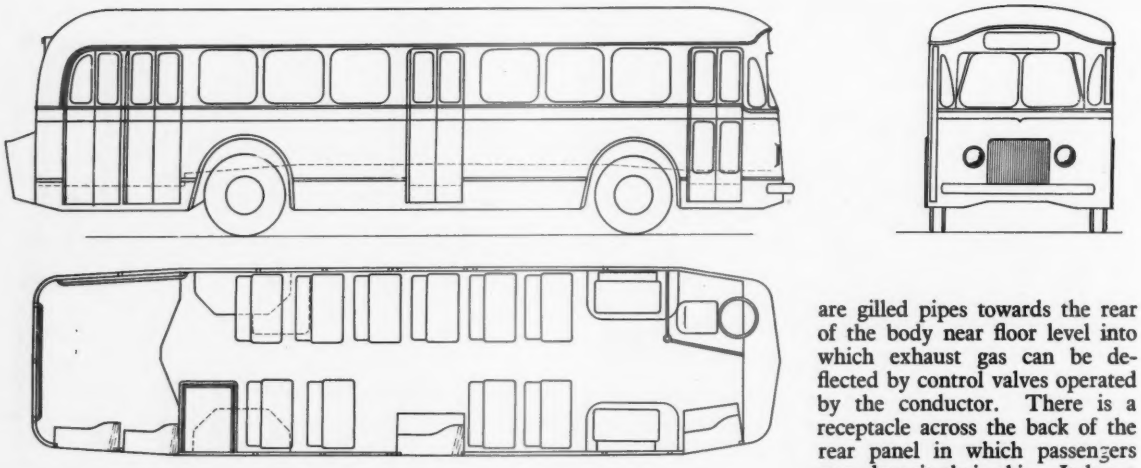
rail and waistrail are of the same open box section as the pillars and are closed internally by the same type of rubber filling.

Below the windows the internal panelling is of Isolite lined with Alfal insulation. The domed roof is of sheet aluminium construction insulated with Isoflex. The maximum headroom over the central gangway is 6ft 3in. Tubular seat frames are used and all seats are in pairs facing forward, except a paired seat on each side facing inwards over the front wheel arches and immediately behind the driving compartment and front exit space. Above the rear wheel arches there are slightly raised platforms each with a paired seat facing forward. That on the left side is backed by a small railed-off parcel space. On the right side is a single inward facing seat for the conductor. This is appreciably raised and can be railed off by hinged bars.

Adjacent to the conductor's seat is an emergency bell push and a stop signal button. Also conveniently placed are the operating levers for the rear and side doors. The conductor collects fares on entry, and does not leave his seat. Passengers wishing to alight indicate their intention by pressing the nearest stop button. This switches on an indicator light. When the bus stops the conductor or driver operates the appropriate door control and as the door opens a large red light is switched on above it together with a similar warning lamp above the driver. These lights are not cut off until the door or doors are closed.

The general floor level is about 2ft 6in above the ground and there are two steps at each door. At the double width rear entrance, rails are provided which separate but do not completely isolate the rear standing

Strømmen bus showing large window area.



Elevation and seating plan.

space from the seated part of the body. The interior lighting is of the concealed indirect type although ordinary bulbs are used. These are positioned within hinged snap-fastened trough sections running along the top of the windows. Above the bulbs in the trough is a translucent strip which serves both to diffuse the light and to intercept any oddments that may tend to find their way in. All light is directed upwards where it strikes the cream enamelled curve of the roof and is reflected downwards. Although the illumination is rather subdued it is evenly distributed and pleasantly restful.

For operation in a country subject to considerable climatic variations throughout the year and with extremely low temperatures in winter, ventilation and heating are of importance. The normal window arrangement includes four half-drop windows on the left side and two half-drops on the right or door side. These windows are mounted in light alloy channel section pans. The fixed windows are directly mounted in the panel openings by rubber glazing. Apart from the ventilation provided by the half-drop windows, air is admitted at all times through ports above the deeply recessed tops of the sloping windscreen panels. These communicate with ducts running beneath the lighting troughs from which the fresh cool air issues.

Rear entry with double doors.

For gentle heating, with additional air circulation, a hit and miss regulator panel in the dash behind the radiator can be opened or closed by the driver to admit more or less warm air into the interior. Incidentally the radiator temperature can be controlled by a driver-operated roller blind between the front grille and the radiator core. The radiator has an area of 173½ sq. in. and the fan is 1ft 11in diameter. For extremely cold conditions there are gilled pipes towards the rear of the body near floor level into which exhaust gas can be deflected by control valves operated by the conductor. There is a receptacle across the back of the rear panel in which passengers may deposit their skis. It has a perforated base so that thawing snow can drain away.

On the road the vehicle evidenced considerable liveliness with very smooth running. With the equivalent of a full load, the acceleration was brisk and cruising speeds of up to 40 m.p.h. were easily held. The engine was not very noticeable apart from a pronounced hiss from the air cleaner which actually projects through the floor under one of the seats. This arrangement provides reasonably dustless air and a fairly constant intake air temperature irrespective of external conditions. A filter with improved silencing characteristics would be desirable.

Weight Distribution

	Front axle	Rear axle	Total
Unladen	3 tons 6 cwt.	3 tons 19 cwt.	7 tons 5 cwt.
Full load	4 tons 13 cwt.	8 tons 6 cwt.	12 tons 19 cwt.

Passenger comfort

Riding was more comfortable than in the conventional British single-deck bus, with relatively greater freedom from shock under light load conditions. The suspension is supple and functions at all times without violent deflections. Luvax-Girling shock dampers are fitted both front and rear.

Acknowledgments are due to both Ingv. Müller of A/S Strømmens Vaerksted and to Leyland Motors, Ltd., for facilities and co-operation in the preparation of these notes and illustrations.

Dimensions: Overall length 32ft 2in, overall width 8ft 0½in, wheelbase 16ft 8¾in, turning circle 82ft. Load capacity 75 passengers, 24 seated, 51 standing: Front 10 × 20in, rear 12 × 20in (single rear wheels).

OVERSEAS ASSEMBLY

A Review of the Methods and Equipment Developed by The Nuffield Organization

TWO economic factors have caused what is known as shipping in the "knocked-down" condition to become of increasing importance to the automobile industry. One of these factors, the unbalance of payments which exists between many countries, is, it is to be hoped, only temporary. The other is without any doubt permanent. Every country with an economy based upon primary products aspires to the introduction of secondary industries. Sooner or later, these countries will make efforts to introduce the manufacture of secondary products.

An obvious method of encouraging the manufacture of such products is to impose protective tariffs to shield the new industries from the competition of established industries in other countries. It is not practicable to build up an automobile industry on such a basis. To begin with, because of the high degree of technical skill that is necessary in both design and production, an automobile industry is only possible in a country already in an advanced stage of industrialization. In addition, the capital investment for even a small but completely integrated automobile industry is so great that a reasonably large assured market is essential. But the only market that can be assured to any industry is the home market, and few, if any, primary prpopducing countries have a sufficiently large home market to warrant the necessary capital outlay.

Industrially undeveloped countries can, however, introduce a certain degree of secondary industry by carrying out assembly functions, since these require a relatively small capital outlay and a lesser degree of technical skill. Because of this, there is an increasing tendency to encourage, and in some countries even to insist upon, the importation of vehicles in the knocked-down condition.

This tendency and the problems involved have been closely studied by the major automobile manufacturers in this country. At present the problems in general are concerned with vehicles that were designed for assembly in the parent factories under conditions of quantity manufacture. However, the importance of the subject is such that in future there is little doubt that knock-down problems will influence the product in the design stage.

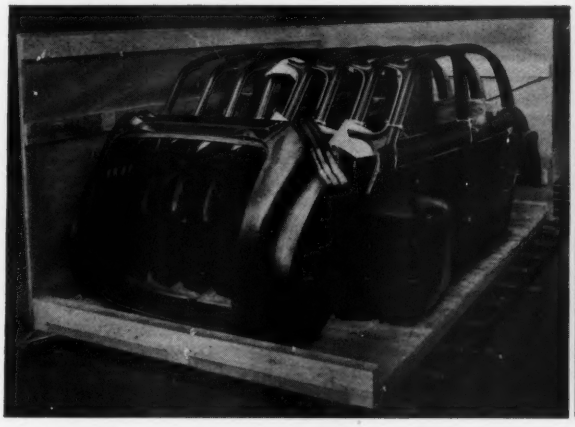


Fig. 1. Side panels and tonneaux ready for packing for overseas shipment.

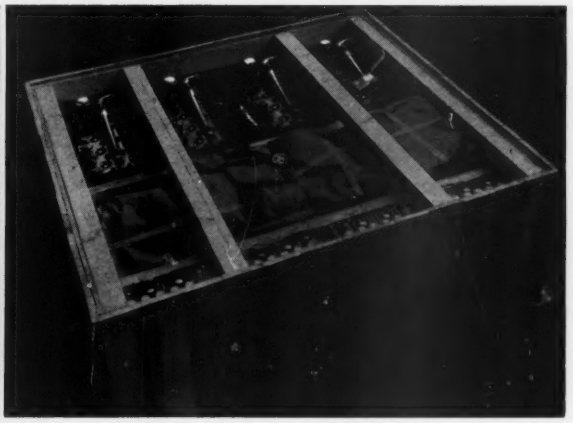


Fig. 2. A case of power units, trim materials and batteries for overseas assembly.

Since overseas assembly introduces some problems that do not occur in the parent factories, and others that differ considerably in degree, it was decided to institute a completely new division of The Nuffield Organization to deal with them. This is known as the K.D. Division, and is an engineering division with full responsibility for all aspects of overseas assembly.

Every class of vehicle manufactured in the various plants of The Nuffield Organization is dealt with by the K.D. Division. That is, the division is responsible for developing equipment and methods for the overseas assembly of the products of Morris Motors, Ltd., Wolseley Motors, Ltd., Riley Motors, Ltd., The M.G. Car Co., Ltd. and Morris Commercial Cars, Ltd.

Essentially, every case involves two main problems. First, the design and lay-out of packages to make the best possible use of shipping space and thus reduce freight charges. Secondly, to develop assembly methods which are economic for the quantities involved, and which at the same time will ensure that the vehicles produced will be of the same standard of quality as those produced in the parent factory.

This question of quality of overseas assembly is, of course, all important. As the assembled vehicle will carry the Company's name, any deficiency on the part of the assembly organization would have an adverse effect on goodwill. Therefore, the primary function of the K.D. Division has been to develop equipment and methods to give quality production at an economic cost even on very small batches.

It is not intended to deal exhaustively with the problems of packing and shipping, although they are important. So far as freight charges are concerned, it is obviously desirable to make the best possible use of shipping space. This, however, will not necessarily be most suitable for the overseas erector since it may involve packing unrelated articles in a single case. For the overseas erector, and particularly when there are only limited unpacking facilities, sequential packing is much the best. Attention must also be paid to customers' requirements when the K.D. engineers determine the degree of breakdown. Typical packs are shown in Figs. 1 and 2.

Assembly manuals

Merely to supply the requisite equipment for overseas assembly is not sufficient to ensure the quality of the end product. The equipment must be used correctly. For this reason, special comprehensive manuals have been prepared by the K.D. Division for each model that is to be assembled. The manual gives ready identification of

Fig. 3. Special deep throat welding gun in use for welding the side panel front to valence.

every part, and arrangements have been made to keep the knock-down specification up to date in the same manner as is a home specification. Grades of change are shown by replacement sheets in different colours. Each manual gives a detailed assembly sequence with a list of the equipment to be used at each operation. Drawings and photographs are used to illustrate points which would not be perfectly clear if only operation sequences were issued.

It has been realized that many of the overseas assembly organizations will have to depend upon relatively unskilled labour. To meet this condition, the manuals have been drawn up in such a manner as to be clearly understandable by people to whom an ordinary engineering drawing would be incomprehensible. Each manual is a tribute to the engineering skill of the K.D. Division.

The assembly of the power and transmission units and the various accessories present little difficulty so long as the sequence as specified in the manual is strictly observed. For these units there are accurately machined surfaces that act as datum points, and since a complete assembly specification is issued to every organization undertaking assembly, there is little danger that trouble will occur with these units.

In knock-down assembly there is no doubt that the greatest difficulty is in connection with the body-shell. This has to be fabricated from a multiplicity of pressings, chiefly by welding. Although the pressings as such are accurate, the datum points are nothing like so positive as the machined datum surfaces of the power and transmission units. In addition, the pressings are of relatively light gauge material, and the welding, if not carefully controlled, can cause considerable distortion with consequent inaccuracy. Since body-shell production is the most difficult function undertaken by overseas assembly organizations, we shall describe in some detail the equipment and methods that have been evolved by the K.D. Division. They illustrate the high degree of technical skill that is exercised to ensure satisfactory overseas erection.

Body-shell accuracy

Before the assembly equipment and methods are discussed, it may be advantageous to describe briefly the means employed by the Nuffield Organization to maintain accuracy of all body-shells. It is based on what is known as a "body reference jig." To the best of our knowledge this jig is peculiar to this organization. For the parent company assembly lines at Cowley, assembled body-shells are received from two sources, Pressed Steel Co., Ltd., Cowley, and Nuffield Metal Products, Ltd., Birmingham. In each of these plants Morris Motors, Ltd., maintain inspectors and a special reference jig for examining body-shells before they are delivered to the Cowley factory for paint finishing and final assembly.

This reference jig is made to engineering limits, and checks all the important dimensions to definite tolerances. The initial location in the jig is taken from the front suspension mountings. From the datum so established, engine mountings, rear suspension mountings, window apertures and valances are checked. When this jig was first proposed, there was a feeling that the permissible tolerances would be so wide as to make the jig relatively useless. Experience has shown that this opinion was completely mistaken.

In point of fact it has been found that despite the inevitable variations in the pressings and in the effect of welding, there is no greater difficulty in working to definite tolerances than there is in machining operations. Any body-shell accepted by this jig is completely satisfactory for the assembly line. Normally, an occasional body-shell is passed through the jib, but as soon as a body-shell shows an approach to a limiting condition, the percentage inspected is increased. This, in conjunction, with an efficient system of floor inspection, ensures that any tendency to depart from the standard is disclosed and corrected before the variations are too great. In developing methods for overseas assembly, the primary consideration of the K.D. Division has been to maintain this standard of quality. That is to say, the equipment had to be such that the body-shells would be produced to the same

Fig. 4. Gun used for welding roof panel to side panel in the drip moulding.

standard of quality as the home production which is controlled by the body reference jib.

Although the standard of quality is the governing factor, processing economies are also of great importance. This introduced its own peculiar problems. The quantities involved in most of the overseas assembly factories are relatively small. Therefore, the capital charges for the equipment must be kept low so that amortization can be completed within a reasonable period and at a reasonable charge per unit produced. The equipment developed ensures the desired standard of quality at an economic cost.

In developing knock-down assembly methods, one of the first problems to be considered is the amount of equipment necessary to carry out the assembly functions efficiently and accurately. So far as body-shells are concerned, much of the necessary equipment is of a standard type that is used in almost every engineering factory, and is therefore also useful for functions other than body-shell assembly. A certain amount of other equipment is, however, unavoidable. It has been kept to a minimum.

Incidentally, there are three knock-down conditions for body shells:—

(a) S.K.D. Complete body-shells, with or without leading.

(b) K.D. Panels in major sub-assembly form.

(c) C.K.D. All pressings straight from the press.

At present (b) is the most common form, but there is a growing tendency towards a wider use of the C.K.D. form. These notes are concerned only with the K.D. form.

The manner in which the K.D. Division has succeeded in keeping special equipment to a minimum is illustrated by the fact that although it now deals with equipment for assembling 10 different models supplied in the knock-down condition, clever design has made it possible to deal with all 10 models on only two main fixtures. One of these fixtures is suitable only for the Morris Minor range of vehicles. The other is used for the Morris Oxford, the Morris Six, the Wolseley 4/50 and the Wolseley 6/50. Some small subsidiary jigs may be necessary for welding certain sub-assemblies, but they are relatively inexpensive.

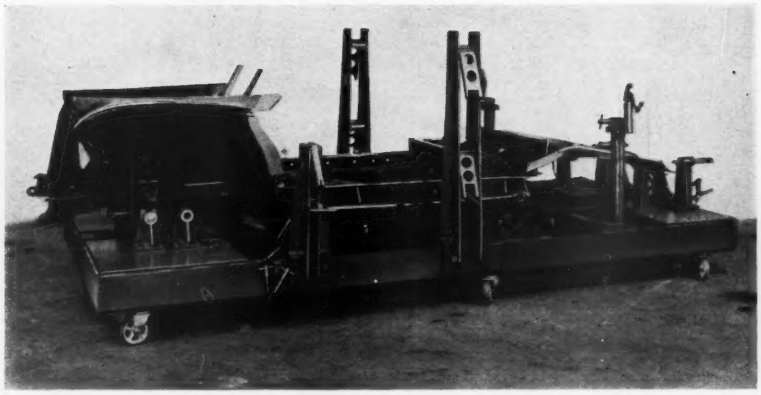


Fig. 5. Main assembly jig for Morris Oxford and Wolseley body shells. A chassis unit is in position.

In addition to the main assembly jig, other equipment called for includes welding guns, clamps, and a truck to carry the assembled body-shell after the work has been completed in the main assembly fixture. This truck has been made adaptable to suit all 10 models. At present only three types of gun welders are required. These and the electrodes used with them have been designed to cover the whole of the spot-welding requirements on both Morris Oxford and Morris Minor body-shells. Two of these guns are shown in Figs. 3 and 4.

To illustrate the manner in which the K.D. Division has dealt with the problems of body-shell assembly, we shall consider the methods and equipment to deal with an output in the order of 30 per week of Morris Oxford, Morris Six or Wolseley cars. For such an output only a single main welding jig is used. For larger outputs, which warrant a greater capital outlay on equipment, two main welding jigs will be employed.

For assembly from the K.D. form the body-shell parts are supplied as follows:—

Chassis complete assembly.
Body side panel assembly.
Tonneau panel assembly.
Roof panel assembly.
Facia and de-mister assembly.
Parcel shelf assembly.
Backlight reinforcement.
Battery box assembly.
Wireless shelf assembly.
Trunk floor support assembly.
Windscreen wiper and motor bracket assembly.
Dash panel.
Filler plate for dash panel.
Strap for wireless shelf.
Squab braces.
Reinforcement for rear post.
Longitudinal support for trunk floor.
Windscreen header panel.
Finisher panel sill assembly.
Stiffener for battery box.

To begin body-shell assembly, the chassis assembly is mounted in the assembly jig as shown in Fig. 5. It is located at six points and supported at two more. The location points are the two front suspension mountings and the four rear spring anchorages. Normal assembly practice is to have the fixed location at the front. In this case the fixed location is at the rear. This practice has been adopted because tll the vehicles for which the jig is designed have a common rear form, but differ at the front. The chassis is also supported at the front of the wheel arch at each side. These locations and support points ensure that the chassis unit is correctly positioned in all directions.

When the chassis is in position, the two side panel assemblies are mounted in the jig. The correct location of these assemblies is of major importance since it governs the width of the body-shell. It is also important that the method of location and clamping is such as to preclude any danger that the body will tend to develop a rhomboidal shape when the welding is carried out.

The main positioning and clamping of each side panel is effected at the front and centre posts by two swinging clamps at each station. As these clamps are integral with the jig and are defi-

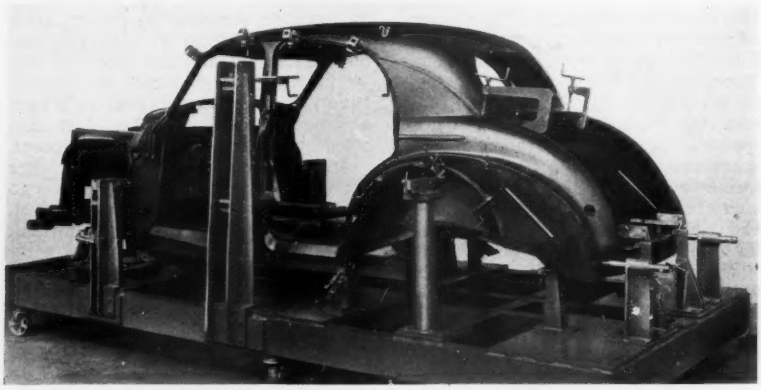


Fig. 6. Chassis, side panels, tonneau and roof panel assembled ready for welding in the main jig.

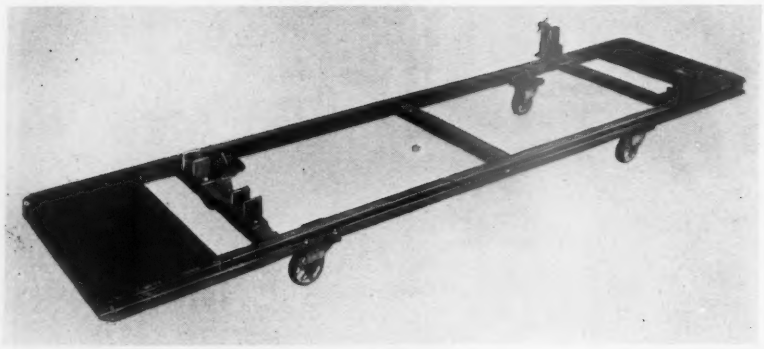


Fig. 7. Body finishing truck.

nitely positioned in relation to the chassis location points, they automatically ensure correct location for the side panels. Experience has shown that the most common faulty conditions encountered in body-shell assembly are, apart from mislocation of the panels, a rhomboidal tendency and a tendency for the side panels to lean towards each other. The rhomboidal tendency has been taken care of by the use of effective location at the forward end of the side panel assembly and effective clamping at the centre post. This centre post clamping also prevents any tendency for the side panels to lean towards each other.

Further location of the side panel in relation to the chassis is effected by means of a special clamp at the top of the rear wheel arch. This clamp not only holds the side panel to the chassis but also gives a correct height location. To complete the mounting, the side panel is clamped to the chassis at the sill and by two locating pins lining up the wing mounting holes to the front post.

At this stage two welding operations are necessary. First, the front end of each side panel assembly is welded to the chassis valance, and secondly the dash panel is welded to the support members on the chassis. It is necessary to carry out this welding at this stage, since at any later stage in the assembly these positions would be virtually inaccessible. While the welding at these stations is proceding, it is, however, possible for a second operator to carry on with the next stage in assembly, that is, the assembly of the tonneau panel.

The tonneau panel is located in relation to both the chassis unit and the side panels. In relation to the chassis it is located by the rear edge of the spare wheel floor while overlapping flanges give the location in relation to the side panels. Further location in relation to the jig itself is afforded by pins which pass through the holes for the rear bumper brackets and register in uprights on the jig. To hold the tonneau securely in position, clamps are attached at the overlapping flange of the rear wheel arch, at the rear and to the side of the chassis floor.

To complete the main assembly of the body-shell, the roof panel is then fitted in position. The main location is taken from the drip mouldings of the side panels while a further location is given by two bolts passing through the trunk lid hinge holes. Specially designed clamps are used to retain the roof panel in the drip moulding and also to attach it to the tonneau panel. When the roof panel is clamped in position, the windscreen header panel and the facia and de-mister assembly are located in position.

At this stage the assembly has reached the condition shown in Fig. 6. It is ready for the main welding operations. A special welding sequence is specified in the assembly manual. There are two reasons for this. The first is to ensure that the welding of one part will not prevent the welding of another part, which might well occur if the sequence were not followed. Secondly, and more important, investigations carried out by the K.D. Division have proved that if the specified sequence is followed both the danger and the degree of distortion are greatly lessened. It is not our intention to follow the welding sequence through in detail. It must, however, be said that it has been carefully planned to give optimum results both for quality of work and economy of time and labour.

Welding jig tests

Every main welding jig for overseas assembly is thoroughly tested by the K.D. Division by building under conditions comparable to overseas assembly a complete body-shell which must pass the body reference jig before approval is given. In addition, the operations are timed to ensure that the specified output can be obtained. Only after the K.D. Division has been satisfied on these points is a jig released for shipment overseas.

When the body-shell assembly is completed, it is transferred from the assembly jig to a specially designed low-bogie truck, see Fig. 7, on which it remains until it is ready for painting. The body-shell assembly rests on a steady at the front and has pin location at the rear.

Body finishing in overseas assembly plants calls for methods that differ greatly from those used in the various assembly plants of the Nuffield Organization. For example, only a very high output, such as that of Morris Motors, Ltd., can justify the installation of a Rotoclip plant for depositing a corrosion-resistant phosphate coating and applying the priming coat and a Rotospray plant for the final painting operations. Nevertheless, finish is an important factor even where the output is small. Therefore, finishing methods for small outputs have been thoroughly investigated by the K.D. Division. The findings of these investigations are incorporated in the assembly manuals, which include complete instructions concerning the materials and methods to be employed to produce a high-quality durable finish at an economic cost.

Although the assembly manuals produced by the K.D. Division of The Nuffield Organization are remarkably comprehensive, there is no doubt that overseas organizations can benefit greatly by arranging for their supervisors to visit the K.D. Division at Cowley. There, the general principles on which the equipment and methods are based can be demonstrated in the K.D. Division demonstration plant. In addition, such visits allow problems that are peculiar to a single organization to be discussed with the engineers of the Division.The value of such visits is being increasingly recognized by overseas erectors.

PANEL TYPE AIR FILTER ELEMENTS

BRITISH Standards Institution have recently issued a British Standard for panel type air filter elements (SP.20). Although this has been prepared to deal with panel type air filter elements intended for use with internal combustion engines and compressors on aircraft, it may prove of value to the road vehicle manufacturer. It is mainly a performance specification and deals with the testing of dry and wet viscous film type elements. In order that filters should be tested under conditions approximating to those encountered in service, dust samples taken from the filters in use on aircraft in different parts of the world were analysed both for constitution and size distribution to obtain a general specification for dust that would be suitable for testing purposes.

It should be noted that the air filter for aircraft use is not called upon to deal with ordinary atmospheric dust, but with the dust raised when aircraft are landing and taking off. As a result the standard includes in an appendix a specification for a test dust which is required to consist of quartz or undecomposed felspar. Methods of carrying out tests are laid down. Copies may be obtained from the British Standards Institution, 24-28, Victoria Street, Westminster, London, S.W.1, price 2s post free.

GAS TURBINES*

A Review of Problems in Their Application to Road Vehicles

By Professor Frank L. Schwartz†

STEADY progress has been made in the development of gas turbines and the question arises whether gas turbines can be successfully applied to vehicle propulsion on a competitive basis with present reciprocating engines. The gas turbine offers several distinct advantages. Comparisons of present and future performance are illustrated in Table I.

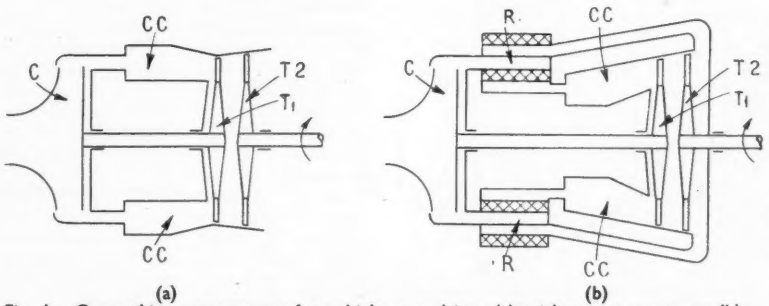


Fig. 1. Gas turbine arrangement for vehicle propulsion, (a) without regenerator ; (b) with regenerator.
C—compressor, CC—combustor, R—regenerator, T1—compressor turbine, T2—power turbine.

Although most gas turbine research and construction has been on units of 1,000 h.p. and larger, there has been some interest in and development of smaller-size gas turbines. Recently the Department of Commerce has announced the development of an automobile propelled by a gas turbine in Czechoslovakia having 60-80 h.p. and designed to burn naphtha, coal gas, hydrogen, acetylene, butane or gasoline.§ (1) Two automotive gas turbines in England have been under development for several years. The Centrax unit is described in (2) (6), and the Rover gas turbine in (3). Although no announcement has been made of an automotive gas turbine in the United States several companies have designed and built small gas turbines. The Boeing gas turbine is a 200 h.p. unit weighing 150 lb. It has a turbine diameter of 10in and a maximum combustion temperature of 1,500 F. (4)

It is difficult to visualize a more complicated propulsion device than a reciprocating engine in which energy from fuel is converted into mechanical work through the effort of pistons, valves, crankshaft and flywheel, then used to drive either a complicated automatic mechanical transmission or a fluid generator which in turn drives a fluid pump and finally a speed reducer at the rear wheels. The only excuse for such a complicated mechanism is that it works with accepted performance.

In comparison, the gas turbine for automobiles offers a simple device consisting of a prime mover and torque converter in a single unit. Only two rotating elements and a speed-reducer are required. Many engineers overlook the fact that the turbine is a torque multiplier. It is one of the few prime movers that are capable of producing high starting torque at low speed. The turbine exerts its maximum torque at zero speed. The gas turbine as proposed for vehicle propulsion (Fig. 1) consists of two separate turbine wheels in which gas flows in series, one to drive a compressor and one to supply power to the vehicle. (5)

The torque requirements for a heavy automobile or medium truck operating at constant speed are shown in Fig. 2 by the dashed curves. These torque requirements are met by the torque produced by the reciprocating engine operating with the gear ratios as shown by the solid curves. The difference in vertical ordinates represents the torque available for acceleration. Fig. 3 shows the corresponding horsepower for the torques of Fig. 2.

The data are based on a 100 h.p. reciprocating engine. Recent data on road vehicle wind and rolling losses (6)

TABLE I

	Present reciprocating engines		Present gas turbines	Future gas turbines
	Manual gear shift	Automatic transmissions	Without regenerator	With regenerator
Miles per gal	15–18	8–14	9–11	16–18
Wt per 100 lb	500	600	150	200
Cost per 100 h.p., $ (high prod.)	250	350	150	200
Advantages	Economical, quiet, simple control.	Smooth operation, quiet, low driving fatigue.	Small size, light weight, low cost, smooth operation, few parts, low-grade fuel, low oil consumption, excellent torque characteristics.	Economical, moderate cost, smooth operation, few parts, low-grade fuel, low oil consumption, excellent torque characteristics.
Disadvantages	Gear shifting, plumbing difficulties, highly refined fuel, oil contamination.	Poor fuel consumption, expensive, plumbing difficulties, complicated mechanism, highly refined fuel, oil contamination.	High-speed reducer, noise, poor fuel consumption	High-speed reducer, noise.

* The American Society of Mechanical Engineers, September, 1949.
† Department of Mechanical Engineering, University of Michigan, Ann Arbor, Michigan.
§Numbers in parentheses refer to the bibliography.

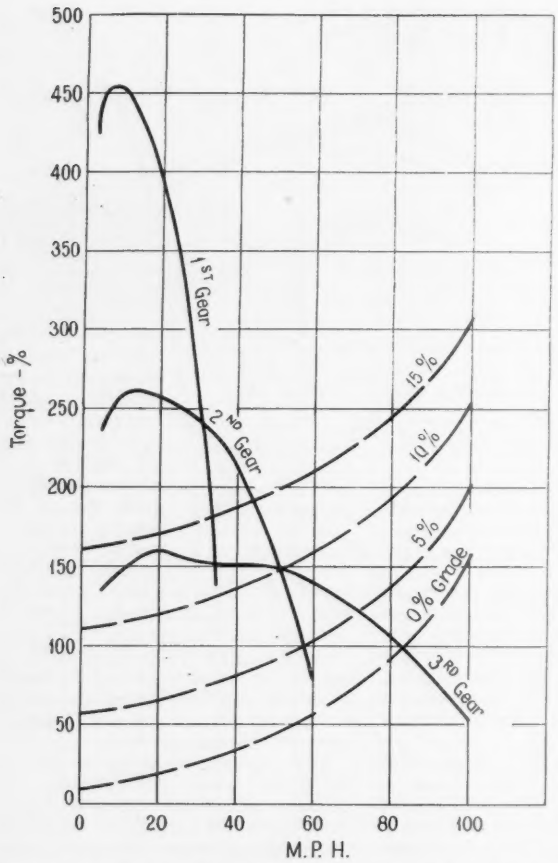


Fig. 2. Torque vs. miles per hour for a reciprocating engine driven car or truck.

show road resistance somewhat less than those assumed in Fig. 2 and 3. The maximum speed of the vehicle represented would be 83 m.p.h. The torque produced by the 100 h.p. reciprocating engine and required on a level road at 83 m.p.h. is taken as 100 per cent. Other torque values are relative. Thus, in first gear the engine is capable of producing 450 per cent. torque at 10 m.p.h. although only 165 per cent. torque is required at the same speed on a 15 per cent. grade. The maximum speed on a 15 per cent. grade in second gear would be 45 m.p.h. and with a torque of 190 per cent.

Fig. 4 shows the performance of a non-regenerative gas turbine designed for the following conditions:

95 h.p. at 83 m.p.h. (same as reciprocating engine).

Temperature at design point—1,500F

Compressor efficiency—80 per cent.

Compressor-turbine efficiency—85 per cent.

Power-turbine efficiency—85 per cent.

Combustion efficiency—95 per cent.

Pressure losses—2.5 per cent.

Pressure ratio of compressor—3.4 : 1.

Inlet air temperature—60F.

On a level road the turbine would propel the vehicle at 83 m.p.h. with a gas temperature of 1,500F. At 68 m.p.h. the gas temperature would be 1,200F. and at 58 m.p.h. 1,000F. The torque multiplication, i.e., the torque at zero speed divided by the torque at maximum speed, is 2.75. Although this is less than the overall torque multiplication of the gear transmission of an automobile in low gear it is more than adequate to negotiate a 15 per cent. grade, Fig. 2.

The power turbine, which is connected directly to the axle through a gear reduction, will operate at speeds from zero to maximum. The compressor-turbine wheel however, operates at nearly constant speed. Thus the compressor and its turbine wheel act as a producer of high-pressure high - temperature gas. Control is effected by reducing the speed of the compressor turbine, thereby reducing the pressure ratio, temperature and air flow through the unit. The speed reduction of the gas producer need not be large to accomplish adequate control. A change in speed of the compressor of less than 20 per cent. will permit operation of the power turbine over its full range of speed.

A study of Fig. 4 will indicate the suitability of the gas turbine to vehicle propulsion. Turbines have characteristically parabolic horsepower curves; efficiency curves are approximately the same shape. With control by gas-producer speed variation, the grade curves fall near the loci of maximum efficiency and horse power. If the design point is selected at the peak of a given horsepower curve, say, one corresponding to 1,500F., the torque multiplication is 2 : 1. By designing the turbine for a blade-speed/jet-velocity ratio somewhat above the ratio of maximum efficiency the grade curves will fall reasonably close to the peaks of the horsepower curves over a range of grade curves and the torque multiplication becomes greater than 2 : 1. Shifting the design point beyond the peak of the horsepower curve causes a small increase in fuel consumption on a level grade but improves fuel consumption considerably on steep grades. Also a slightly larger turbine is required.

TABLE II

Windage loss, H.P. $= \dfrac{(D)^5\ (N)^3}{(10)\ (1000)} \Big/ 285 \times 42$

N	D = 4″	8″	12″
5000	0·0001059	0·00339	0·0242
10000	0·00085	0·0272	0·206
20000	0·00678	0·217	1·64
23800			2·82
30000	0·0229	0·735	
35800		1·76	
40000	0·0543		
71600	0·309		

Underlined values correspond to a pitch line velocity of 1,250 f.p.s.

A vehicle requires maximum power and maximum torque for less than 1 per cent. of its operating life. To obtain high torque output for short intervals the gas temperature may be raised above the design temperature. If the gas temperature is raised to 1,800F. the torque multiplication becomes 3.55 to 1. This procedure is analogous to supercharging aircraft reciprocating engines during take-off

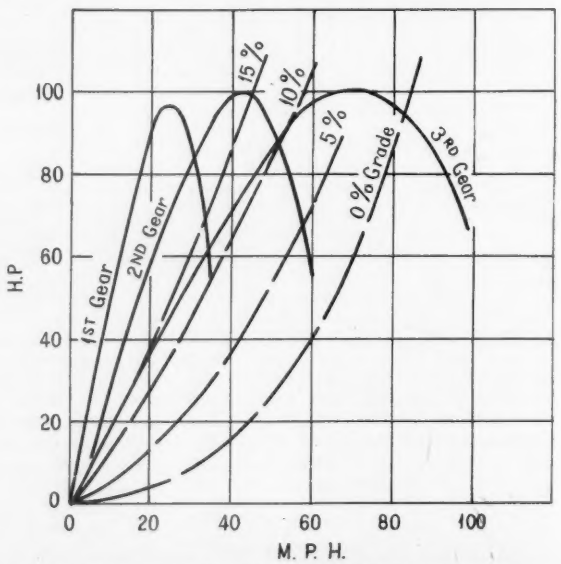


Fig. 3. Horse-power vs. miles per hour for a reciprocating engine driven car or truck.

wherein a time limit is set during which the intake manifold pressure may be raised above normal.

Fig. 5 illustrates similar data for a gas turbine using a regenerator having 75 per cent. effectiveness and allowing 0.5-psi pressure drop on the air side and 0.5-psi pressure drop on the gas side. Fig. 6 and 7 show specific fuel consumption and miles per gallon for each cycle. Without regeneration the fuel consumption is poorer than that of present reciprocating engines using gear transmissions and hand shifts. However, when compared with the fuel consumption of heavier cars and trucks using automatic transmissions the fuel consumption for the non-regenerative gas turbine Fig. 6 is comparable.

With regeneration the fuel consumption is materially improved. The mpg reaches values comparable with many automobiles and light trucks now in operation. The principal advantages of the gas turbine drive are its excellent torque characteristics eliminating automatic transmissions, freedom from plumbing difficulties, no need for anti-freeze, low consumption of lubricating oil, light weight, small number of moving parts, and ability to use fuels that do not require a high degree of refining, i.e., low octane number, etc.

There are many problems to be solved before the gas turbine becomes a practical machine that is competitive with the reciprocating engine for vehicle propulsion. Among these are component efficiencies when parts are made small, high-speed gearing, noise, effect of dirt deposits and low-cost production techniques. However, progress to date on gas turbines does not indicate that these problems cannot receive adequate solutions.

Consider the losses in a turbine. These losses may be divided into nozzle, blade, leakage, windage, leaving losses and bearing friction.

Considerable experimental work has been done on nozzles and some of the results have been published (8), (9). If the aspect ratio (radial height divided by throat width) is kept high to prevent end boundary layers and end parasitic flow patterns from causing a high disturbance in the nozzle flow and the nozzle velocity does not exceed 20 per cent of the local sonic velocity, high nozzle efficiency may be maintained. Nozzle efficiency of 94 per cent and better can be had with proper design.

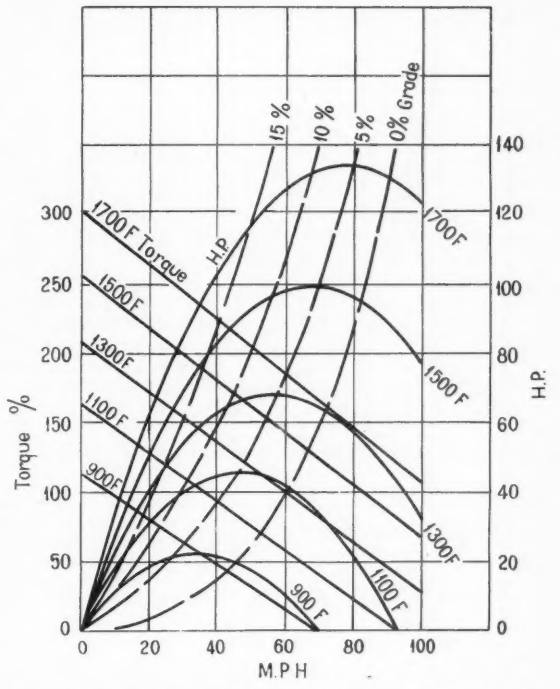


Fig. 5. Horse-power and torque vs. m.p.h. for a dual gas turbine with 75 per cent effectiveness regenerator.
———Horse-power and torque for gas turbine.
— — —Horse-power required.

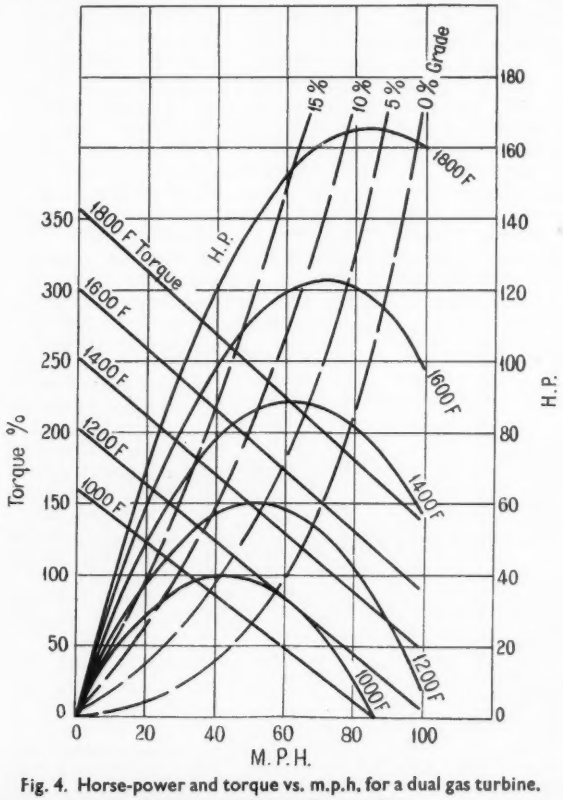


Fig. 4. Horse-power and torque vs. m.p.h. for a dual gas turbine.
———Horse-power and torque for gas turbine.
— — —Horse-power required.

The most serious loss in small turbines is the blade loss. The blade loss is due to secondary flow and profile friction. The duct aspect ratio, (width/depth), determines the proportion of the total flow affected by secondary flow in the blades. If blades are made very short the secondary flow causes a large loss. The greater the width of the duct the smaller is the proportion of the fluid affected by the secondary flow. Every attempt must be made to keep the duct aspect ratio as large as practical. This results in wide blades closely spaced. The efficiency of the blade in the central portion is essentially constant so that if the secondary flows at the ends of the blades can be kept low good blade efficiency can be approached; long blades will tend to reduce the percentage of secondary flow loss. Also, the shape of the blade passage affects the secondary loss. Blade sections which permit the turning to take place before the fluid is accelerated gives best results and by turning the fluid at low velocity the secondary losses are reduced.

TABLE III

Nozzle	4·0	per cent
Blade	5·0	,,
Leakage	1·5	,,
Windage	0·5	,,
Leaving	2·5	,,
Bearings	0·25	,,
Accessories	1·0	,,

Profile losses are functions of the velocity ratio, pitch-width ratio, Mach number and geometry of the section. More will be known about this loss as more blade-pack test data become available. A general rule which shows qualitatively the value of this loss is Blasius's formula :—

$$\text{Blade loss} = \frac{\text{constant}}{\sqrt[4]{\text{Re}}} \times \frac{L}{D} \times \text{kinetic head}$$

where Re = Reynolds number $= \frac{\rho VD}{\mu}$

The $\frac{L}{D}$ ratio (width of blade/hydraulic

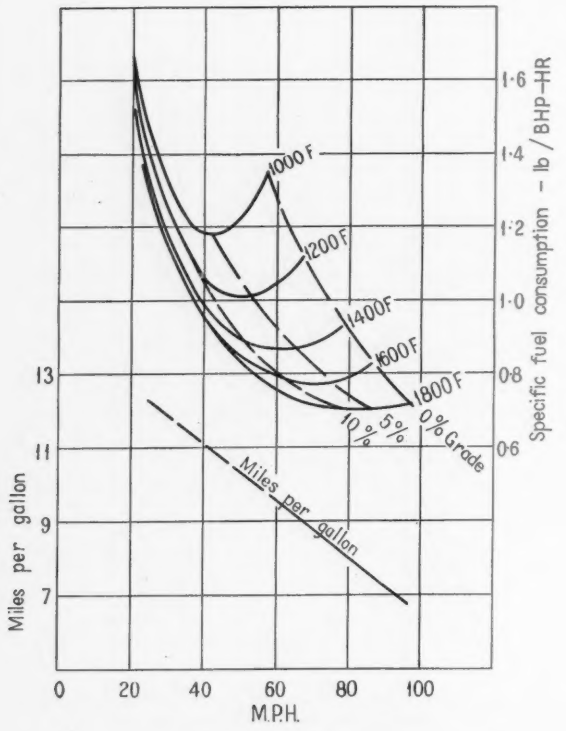


Fig. 6. Specific fuel consumption and miles per gallon vs. miles per hour for dual gas turbine cycle.

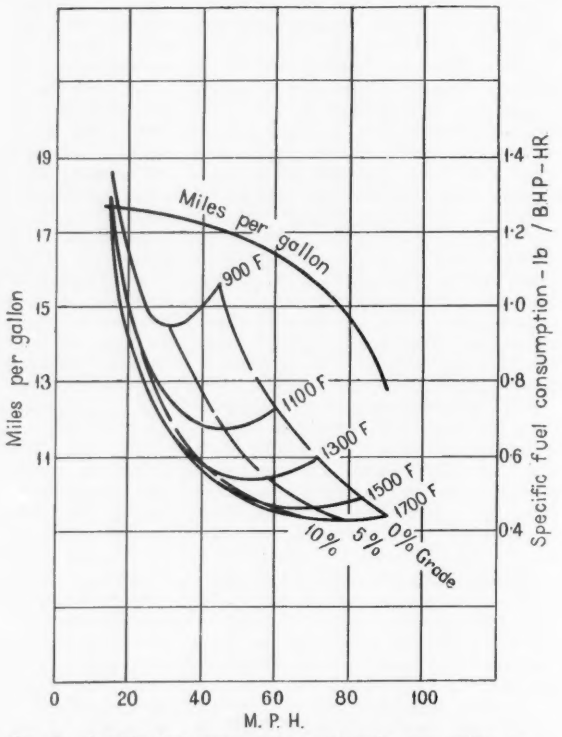


Fig. 7. Specific fuel consumption and miles per gallon vs. miles per hour for dual gas turbine cycle with 75 per cent. effectiveness regenerator.

diameter of blade passage) may be kept approximately the same in a small turbine wheel as in a large wheel, but the blade loss varies inversely with the fourth root of the Reynolds number, which in turn decreases as the size of the turbine wheel decreases. The density ρ, the velocity V and the viscosity μ need not be appreciably different in a small wheel than in a large wheel. However, the hydraulic diameter decreases with a decrease in turbine diameter and the blade losses increase.

The leakage loss depends on the percentage of leakage area, and is influenced by the velocity ratio. Fig. 8 shows the clearance area/blade-annulus area plotted against wheel pitch diameter for several ratios of blade height L/pitch diameter D and several clearances c. For wheels over 12in in diameter the clearance ratio is only a few per cent, but at wheel diameters of 4–8in the clearance area can lead to excessive losses. The curves show the importance of keeping clearances small and L/D large. Since the total creep in a small wheel is less than in a large wheel one can afford to have smaller clearances in a small wheel. A large L/D not only decreases the clearance loss but also decreases the blade losses. On the other hand the blade stress and the amount of twist in the blades increase with L/D.

Small turbines require high rotative speeds and one is led to believe that high speeds incur excessive windage losses. Stodola (10) and Kearton (11) show the windage loss to be proportional to the fifth power of the pitch diameter D in and the third power of the r.p.m. N

$$\text{Windage loss} = \text{Constant}\,\frac{D^5N^3}{v}$$

where v = specific volume

Table II indicates that windage horse-power does not exceed about 1 per cent for a 150 h.p. turbine and is negligible for turbines less than 12in diameter. The underlined values correspond to a pitch-line velocity of 1,250 f.p.s., which is a high limit on blade speeds. Large wheel diameters are more conducive to high windage loss than high r.p.m. In turbines of very small size there is an urge to use partial admission and thereby increase the size of the blade passages and reduce blade losses. However, the pumping loss in the closed-off blades usually amounts to more than the blade loss and partial admission should be avoided.

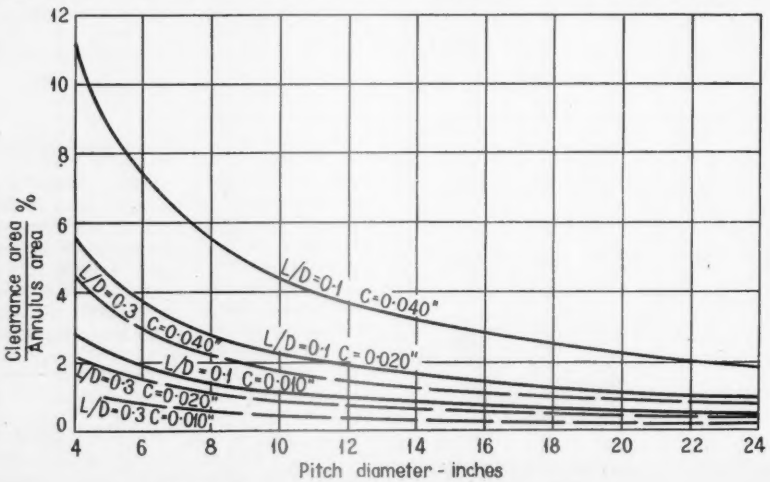


Fig. 8. Per cent. clearance area vs. pitch line diameter.

The leaving loss in jet-propelled planes is not a total loss, for it can be used in the nozzle to aid propulsion. In multistage turbines the leaving loss from one stage can be

partially recovered in the succeeding stage. In a single-stage turbine for producing shaft power the leaving loss can amount to a high percentage loss. By keeping the direction of the exit velocity parallel to the turbine shaft and using a low axial velocity the leaving loss may be kept to small values. Small axial velocities lead to longer blades, thereby increasing stresses but decreasing blade losses. Fig. 9 shows the equivalent B.T.U. loss per lb of gas, and the corresponding horse-power loss for a flow of 1 lb per sec, plotted against leaving velocities. If the leaving velocity can be kept at values of 200–400 f.p.s. the leaving loss will be small. Bearing friction is negligible in that only four ball or roller bearings are needed. Accessories requiring power from the turbine include a fuel pump, oil pump and governor.

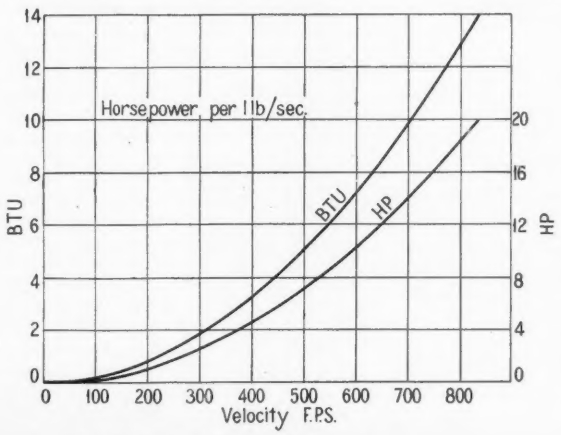


Fig 9. Leaving loss vs. leaving velocity.

To obtain an 85 per cent turbine efficiency, the 15 per cent of losses would need be distributed about as in Table III :

For small turbines the radial inward-flow design offers some advantages over the axial-flow type. The leakage loss in a radial turbine is only about one-eighth of that in an axial turbine. Efficiencies of 85 per cent have been reported and variable nozzle angles become a possibility. (12)

Regeneration offers considerable improvement in overall efficiency. However, when surface-type shell-and-tube or plate-type exchangers are used the size and weight as well as cost remove the lustre from the theoretical advantages. Small size requires high gas-film coefficients of heat transfer, which in turn are accompanied by high gas velocities. But high gas velocities incur high pressure losses and the gas-turbine cycle is very susceptible to pressure losses. (13)

The rotating regenerative type of heat exchanger, in which fine wire elements are heated by hot exhaust gas for 180 deg and then cooled by compressed air on its way to the combustor, permits high effectiveness with low pressure losses. The Ritz heat exchanger, of this type, received considerable attention in Germany and England. The major problem in this type of heat exchanger is the development of a sliding seal which will have a long life at high temperature. Progress on this type of heat exchange has been reported in England. Fig 10 shows relative areas for surface-type heat exchangers versus effectiveness. The area and bulk required for heat exchangers much above 50 per cent effectiveness becomes prohibitive. However, the Ritz heat exchanger is capable of 90 per cent effectiveness without abnormal weight.

Small turbine wheels made of high-temperature alloy can be cast in one piece with the blades integral with the wheel. Wheels of 9in O.D. have already been successfully cast and could be cast on a production basis. Likewise, centrifugal compressors having double shrouds can be cast of aluminium in a single piece. The

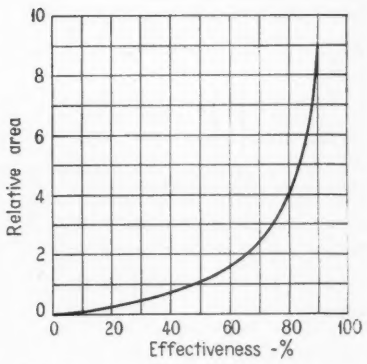


Fig. 10. Surface heat exchanger area vs. per cent. effectiveness.

TABLE IV

Mfg.	Application	Design temp.	Air Flow	Press. ratio	Compressor			Turbine			R.p.m	H.P. or Thrust	S.F.C.	
					Type	Eff.	Dia.	Type	Eff.	Dia.				
		F	lb/sec			%	in		%	in			lb/ h.p.-hr	
Boeing	Turbo prop.	1,500	3·5	3·0	Cent.	76	8·5	Axial	80	7·5	36,000	H.P. 160	1·2	Compressor Turbine
								Axial	75	10·0				Power Turbine.
Flader	Classified	1,250	15·4	2·85	Class.	Class.	15·75	Class.	Class.	Class.	28,200	Thrust, lb 770	1·64	Take off
		1,170									26,800	Thrust, lb 700	1·65	Continuous
		1,040									19,800	Thrust, lb 350	1·96	Idling
Solar	AAP 80 Aircraft APU	1,500	2·25	4·2	Axial	80	7·5	Axial	81	8·9	24,000	H.P. 140	0·925	
	MAP 400 Navy APU	1,500	6·1	4·71	Axial	83	10·25	Axial	83	12·0	20,000	H.P. 415	0·90	

Additional units have been built but the data are not available because of classification restrictions.

GAS TURBINES (Continued)

manufacturing of the few parts required in a small gas turbine could be performed at very low cost on a production basis.

The data presented in Figs. 4–7 are based on values which appear to be attainable at the present time. Gas turbine research will improve the situation. Development work in metallurgy, air-cooled blades, supersonic compressors of high efficiency, improved turbine-blade forms and high-effectiveness heat exchangers will lead to higher efficiencies and lower fuel consumption.

The 80 per cent compressor efficiencies and 85 per cent turbine efficiencies assumed are readily attainable at the present time in large units. Small compressors of 79 per cent efficiency have been built and small turbines of 81 per cent efficiency have been tested. Considering the reduction in losses between the turbine and vehicle wheels which the turbine characteristics provide over the transmissions required for reciprocating engines, the gas turbine for vehicles offers competitive advantages. With further development, temperatures of 2000F and air-cooled blades will increase the performance to very attractive values. Much of the success of the gas turbine for vehicle use will depend on further advances made in the art.

Some of the problems involved in the successful application of gas turbines to vehicle propulsion are noise and vibration, cost, control, starting, exhaust-gas dissipation, air consumption, stresses, fuel system and weight. The noise problem is particularly troublesome. Much of the noise is airborne and originates in the compressor. However, it is high in pitch and therefore easy to absorb. Sufficient research should find a solution of the noise problem. Vibration is not harmful. Since there are only two high-speed rotating parts in perfect balance, vibration does not become objectionable.

Cost has already been discussed and on a production basis should be low. The high cost of present turbines is due primarily to development costs. The cost of the engine for a vehicle is a small part of the cost of the vehicle so that whether the cost of the engine is one half or twice present costs does not affect the vehicle cost to a great extent.

Control of gas turbines over a range of speed and load is not easy. Adequate controls are becoming more complicated and costly. Fuel nozzle research to date has brought on complications rather than simplification. However, the gas turbine for a vehicle needs a single control for the fuel combined with a temperature limiter and an overspeed control. Small gas turbines already in operation may be brought from a starting state to full load in fifteen seconds and after starting the compressor turbine the power turbine may be accelerated from stall to full speed in five seconds. An electric starter is required to accelerate the compressor turbine to about 25 per cent of rated speed. Ignition is required for only a few seconds at starting. Starting can be made easier by opening a bypass between the two turbine wheels so the compressor turbine can use the full pressure ratio during starting.

The gas turbine uses about seven times more air through the machine than does an equivalent reciprocating engine. Therefore, there is seven times as much exhaust gas. However, the exhaust gas from the gas turbine is cooler than the exhaust gas from a reciprocating engine and this is especially true if a regenerator is used. Including the air requirements for engine cooling of reciprocating engines, the air requirements for the two types of engines are about equal. Hot gas flowing from the rear of a gas-turbine-propelled vehicle is no different from that of a conventional automobile.

The turbine blades are subject to high stresses and a mechanical failure of a blade presents disastrous results. Such troubles can be avoided by adequate design and sufficient preliminary testing with perhaps new parts installed at determined intervals. Alloys having one tenth per cent creep in 3000 hr under a stress of 30,000 psi at 1350F are available at present. If blade-vibration fatigue is eliminated these alloys are adequate for the life of a vehicle.

Gas turbines can effect a 50 per cent saving in weight of the engine. A saving in one lb of engine is accompanied by the saving of another lb in chassis weight so that material saving in weight and cost of a vehicle is realized.

The reciprocating engine has received the attention of thousands of engineers and millions of dollars spent in a combined effort to make it a reliable, practical and acceptable efficient machine. If only a small portion of this time, effort and money were expended on gas-turbine development a highly efficient, reliable machine with improved performance could be made at reduced costs.

Table IV lists some of the small gas turbines which have been built to date. In nearly all cases these machines were the first effort of the companies building them and at a time when the progress in gas-turbine research was in its infancy.

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FUNDAMENTAL RESEARCH

AT the Massachusetts Institute of Technology two new developments concerned with automobile engineering are now being undertaken. In the Sloan Laboratory for Aircraft and Automobile Engines the effects of size changes in single-cylinder engines are being explored in an attempt to establish laws of similitude that may be of general use. The ultimate aim is to develop a method whereby a small or simplified working engine model could be used to study the characteristics of a large engine before it is built. To this end three geometrically similar engines of different sizes have been built on which work is proceeding to show how air flow and stresses within the engines may be compared. Other factors, such as friction, lubrication and heat losses, are also being studied.

In the aeronautical engineering department an improved high-performance engine indicator is under development. Although this instrument applies commonly used strain gauge principles for measuring pressures inside an engine, it is claimed that the defects of existing devices are largely overcome by the use of a curved instead of flat pressure recording drum and by refinements in the application and instrumentation of the strain gauge itself. It is claimed that the instrument is accurate to within 1 per cent over a wide range of temperatures and that while large variations of cylinder pressure can be measured, it is sensitive to very rapid pressure changes. Consequently, it may make possible detailed studies of rapid variations and oscillations of pressure within an engine.

SPECIAL PURPOSE BODIES

A Review of Production Methods Employed by Dennis Bros., Ltd.

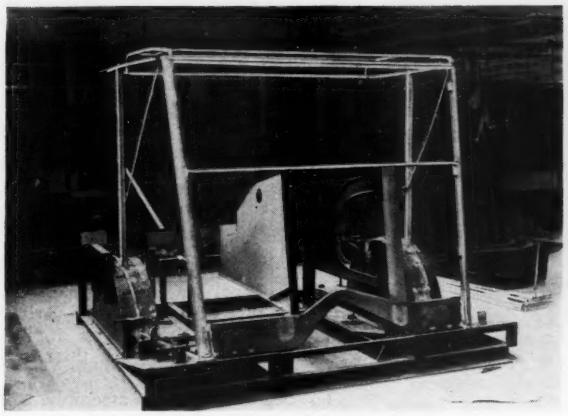


Fig. 1. Assembly jig for all-metal cab frame.

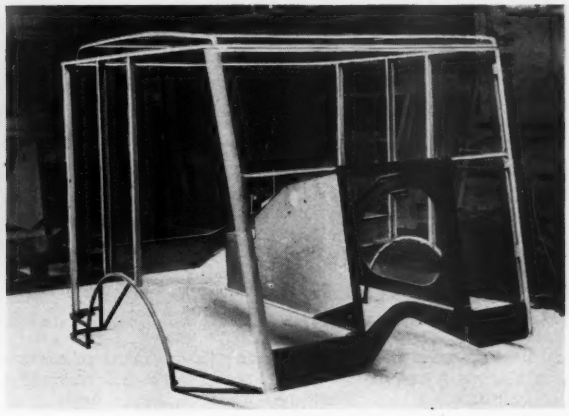


Fig. 2. Assembled frame for all-metal cab.

SINCE it is the policy of Dennis Bros., Ltd., Guildford, to produce complete vehicles to customers' specifications, their production problems in body manufacture differ considerably from those of undertakings of comparable size producing only a range of standardized vehicles. In addition, not only does this organization produce a wide range of vehicles, but there are also considerable differences in the vehicles belonging to the same class.

Some of the differences are merely expressions of personal preference and are not easily justified. Many, however, are governed by operational efficiency. For example, this organization has for long specialized in the manufacture of municipal refuse collectors, and every town has its own peculiar problems that must be considered if optimum operating efficiency is to be obtained. To take only one factor, the width of the vehicle is frequently governed by the width of the streets and alleyways in which it will have to be manœuvred. This means that while some towns can operate vehicles of maximum legal width, others must have narrower vehicles.

Although this deliberate policy of designing bodies to suit specific requirements means that there is little standardization of finished product, clever design and process planning have led to a much greater degree of standardization in production than appears possible at first sight.

Factory organization

In organizing the factory for body production, product grouping has been employed as far as the actual assembly is concerned. So far as is economically justified, each of the body departments is completely self-contained. Practically the only exception is that to avoid duplication of expensive machinery and to make the fullest possible use of the machines, all cutting to size of steel sheets is carried out in one department, irrespective of whether the sheet is to be used for all-metal or for composite bodies.

There are three distinct body departments:—

(1) All-metal bodies.

Fig. 3. Dennis " Pax " tanker with all-metal fireproof cab.

(2) Paxit bodies, a special form of all-metal body.

(3) Composite bodies.

In addition to the three distinct body departments there are two centralized departments:—

(a) Radiators and petrol tanks for all vehicles.

(b) Body painting and finishing for all vehicles.

All-metal bodies

Owing to the multiplicity of bodies produced and to the fact that production is on a batch rather than a quantity basis, it is not economically feasible to make the same use of drawing presses as in a factory producing only a few types of bodies in fairly large quantities. There are, of course, certain components that are common to many of the bodies. For these it is economical to prepare and use expensive press tools. In addition, it has been found possible to draw certain other parts on presses, although the actual number required for any one model is not sufficient to warrant heavy tool costs.

Although the maximum possible economic use is made of drawing presses, five presses suffice for all the blanking and drawing. These presses are:—

One 250 tons.

Fig. 4. Dennis " Paxit " refuse collector.

One 100 tons.
One 25 tons.
Two 15 tons.

For work that cannot be carried out economically on a press, guillotines, nibbling machines, press brakes and bending rolls are employed.

There are two components in particular which illustrate the manner in which a single pressing may serve for many different vehicles. That is, a number of final forms can be produced from one single form of pressing. Consequently, the total production is sufficient to warrant the heavy initial cost of press tools, and the runs are lengthy enough to offset the costs entailed in setting up the press.

For example, one form of pressing is used in the production of 14 different radiator shells. The only difference between any two forms of shell is in length. As drawn in the press, the forms of top and bottom of the shell are produced as a unit. The drawn form is then pierced and split to produce a shell top and bottom. These parts are welded on to side pieces, formed to shape on a press brake, to form a radiator of the desired length and of the distinctive Dennis type.

Similarly with the roof corner pieces for all-metal cabs of municipal refuse collectors. For these vehicles there are ten different cabs, but two pressings, one right-hand and one left-hand, suffice for the roof corner pieces of all ten.

Once again the variation is merely a matter of length. For the longer cabs, the pressing is cut and a lengthening piece is welded in. For shorter cabs, a section is removed and the ends are then welded together to make a complete roof corner.

Despite design and production ingenuity it is not possible to have more than a small proportion of each all-metal cab produced from conventional pressings. For this reason, great use is made of press brakes. As far as possible, parts which cannot be produced by conventional press methods because the quantities are too small are designed for production on a press brake or in bending rolls. This calls for a higher degree of operational skill and therefore higher labour costs than normal press operation. It is, nevertheless, still the more economical.

The manner in which many different types of all-metal bodies are produced economically and accurately may be illustrated by the methods used in the manufacture of all-metal cabs for municipal refuse collectors. Ten different types of cabs are produced on one assembly line. They all start on the same assembly jig. This jig is so constructed that on changing from one cab to another the necessary adjustments to the datum points can be made quickly. The jig is illustrated in Fig. 1.

Essentially, the cab comprises a bottom frame, front, centre and rear posts, front and rear presses cross-members for the engine mounting, a top frame and a sheet metal skin. The top and bottom frames are fabricated from 16-gauge, $\frac{3}{4}$in square tube. To produce these frames, the various lengths are bent to form on a pipe-bending machine and are then arc-welded in a jig.

To begin the main assembly, the bottom frame is mounted on the assembly jig in relation to the datum points and clamped in position. The arc-welded screen aperture sub-assembly, comprising special section front posts that are produced on a press brake and lateral stiffening sections, is then placed in position and clamped to the bottom frame, as are the centre and rear posts. Following this, the top frame is bolted to all posts and a diagonal stiffener is clamped to the front post and the centre post on each side.

At this stage the front and rear cross-members are clamped in position. Incidentally, these members are produced by conventional press methods. One pressing is used for each member

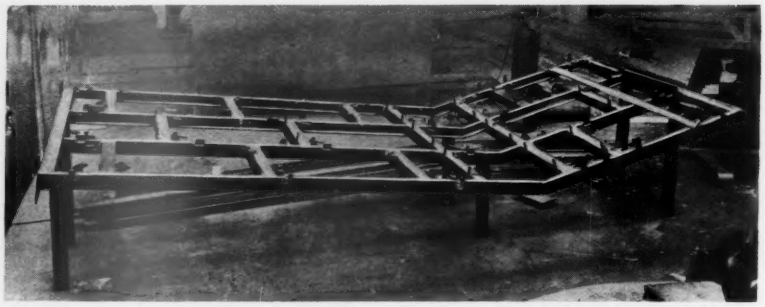


Fig. 5. Welding jig for the steel sub-frame of a " Paxit" vehicle.

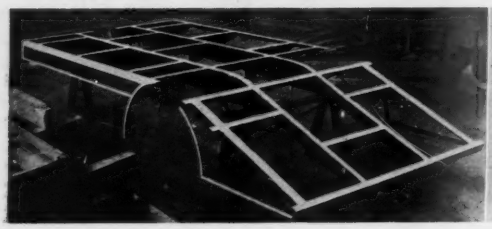


Fig. 6. Assembled steel sub-frame for the "Paxit" vehicle.

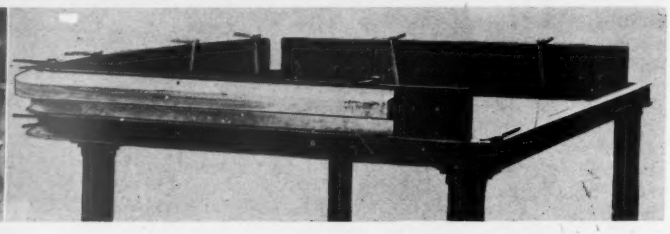


Fig. 7. Drill jig for the body arches of the " Paxit."

for all cabs. It is cut and re-welded for narrower cabs, and for wider cabs an extension piece is welded in. To complete the assembly on the main jig, the engine and gear box cover plates, which are the same for all cabs, are bolted to the cross-members. When the assembly is complete, and before it is removed from the jig, the posts are arc-welded to the frame and cross-members. The assembly as produced in the jig is shown in Fig. 2.

After the welding is completed, the frame is removed from the jig and transferred to the next station. Further jigging is not required, since the assembly is sufficiently strong to maintain form throughout the remaining operations. At the second station the roof corner panels are fixed in position and the side panels are riveted to the frame. The roof panels are also attached at this station. These panels are supplied to the assembly line cut to size, but not formed to shape. They are bent to form against the assembled

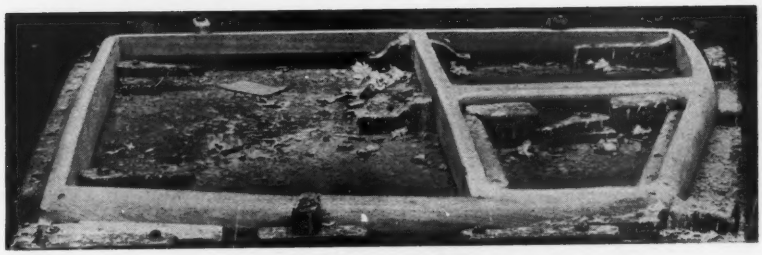


Fig. 8. An assembly jig for composite cab frame components.

Fig. 9. Checking fixture for composite cab frames.

frame. The doors are produced as complete sub-assemblies at another section. At cab assembly they have only to be hung in position. To complete the cab assembly, the windscreen, the side windows and the radiator are fixed in their respective positions, and at the end of the line the cab is ready for mounting on its chassis. A Pax tanker with an all-metal fireproof cab is shown in Fig. 3.

"Paxit" refuse collector

The most recent development in refuse collectors is the Dennis vehicle known as the "Paxit" mechanically operated, rear-loading vehicle. It is illustrated in Fig. 4. Loading is carried out at the rear. The loading height is 4ft 6in from ground level. It is interesting to note that although the vehicle has a low loading height, it is equipped with 34×7 heavy duty tyres. The chassis employed is the standard Dennis Pax 11ft 6in wheelbase, forward control model, and the fully enclosed driver's cabin is streamlined. It has accommodation for six loaders in addition to the driver.

Essentially, the body comprises a steel sub-frame that is mounted on the vehicle chassis to carry the body proper, which is fabricated from a special aluminium alloy. The steel sub-frame is a completely welded fabrication. It is fabricated by conventional arc-welding methods in the jig shown in Fig. 5.

To facilitate welding, the jig is mounted on a welding manipulator which allows the whole of the welding to be carried out from a minimum number of positions. After the sub-frame fabrication is completed, the top of the assembly is painted with chromate jointing compound to prevent any metal-to-metal contact between the steel sub-frame and the aluminium superstructure. If this precaution were not taken, electrolytic action would be set up and corrosion would be initiated. A sub-frame ready for assembly on the chassis is shown in Fig. 6.

The body proper is built throughout of special aluminium alloy. This material is non-corrosive and, as it is used in the fully heat-treated condition, it has exceptionally good resistance to wear. In addition, the use of this material gives a vehicle of minimum unladen weight. It also eliminates the necessity for painting, which leads to a considerable saving in maintenance costs over the life of the vehicle.

At the rear, the body is in the form of a hopper that houses an hydraulically operated loading plate. This plate covers the full cross-sectional area of the hopper below the loading level. The loading plate runs in two brass guides on the centre and is controlled on its outer edges by two brass slippers. Leather scrapers are fitted to protect the inside panels against wear from the refuse. The panels in contact with the leather scrapers are bolted in position so that they can be removed easily as and when wear occurs.

In loading, the contents of the bins are tipped over the loading line to the forward side of the loading plate. When this portion of the hopper is full—it takes approximately 20 bins of refuse—

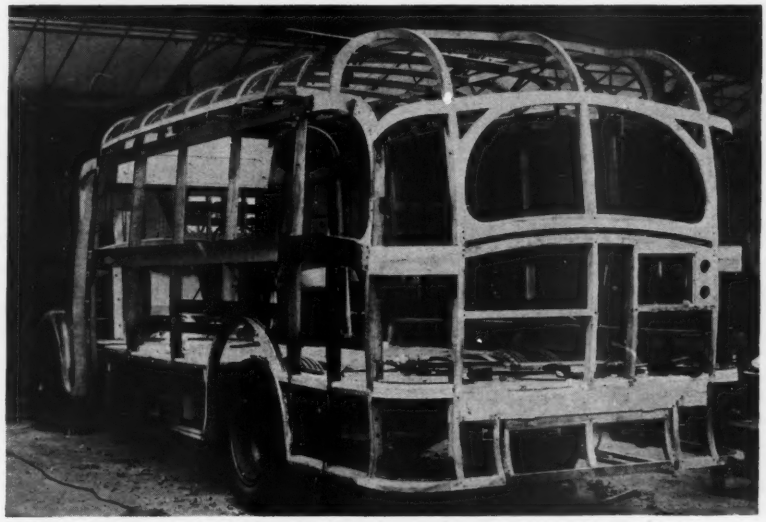


Fig. 10. Framing a composite body for a commercial passenger vehicle.

Fig. 11. Dennis bodywork on a Falcon chassis.

the loading plate is brought into operation to push the refuse into the body proper. This operation is repeated until the body is filled to roof level. Twin hydraulic rams are used for elevating the hopper, and the opening is such as to give a clear slip-way when the body proper is elevated to an angle of 45 deg. for tipping the contents. Power-operated hydraulic tipping gear is fitted, with the rams situated at the forward end of the body.

In the main, the body is of all-riveted construction. Where bolts are employed for fixing brackets which are to be readily detachable to facilitate servicing, they are Sherardized to prevent metal-to-metal contact between the steel bolts and the aluminium alloy structure. This care for detail is typical of Dennis production.

Extruded sections of special aluminium alloy are used for the frame arches. They are supplied bent to the correct form. All the necessary holes in the frame arches are drilled with portable pneumatic drills in jigs of the type shown in Fig. 7. While the main frame members of the body are supplied and used in the fully heat-treated condition, the sheets for the roof panels are supplied and used in the solution heat-treated condition.

As the frame arches are supplied bent to the correct form, there is no great need for machinery in this department. In fact, the only machines used are a Midcyl saw and a Wadkin drilling machine. The Midcyl saw has a universally adjustable head. It is used for cutting rolled sections to length and at correct angles for mitred joints. The Wadkin high-speed drill has a retractable radial arm. It is used for drilling rivet holes in the aluminium alloy sheets.

The hydraulic gear for the Paxit vehicle is simple in design. It comprises a single pump power take-off unit bolted to and driven from the side of the road gear box of the vehicle. The oil reservoir is contained within the end of the tipping ram assembly. In operation, oil is pumped via a release valve to a simple control box which distributes the oil to the selected ram. Selection is made through a single lever fitted inside the driver's cabin. It is controlled by a gate having three neutral and six operating positions. A relief valve cuts-off at a maximum working pressure of 2,000 lb per square inch.

Fig. 12. Dennis fire-fighting vehicle to Ministry of Works/Home Office standard design.

Composite bodies

The composite body department comprises several sections. They are: a sawmill, cab assembly, frame assembly, final assembly, panelling and a trim section. As with the all-metal bodies, many different types are produced, although in general they fall into three classes, commercial passenger, fire fighting and municipal. In connection with fire engines it is interesting to note that there is now complete standardization so far as engines for local authorities in this country are concerned. Dennis Bros., Ltd., still supply fire engines to many overseas authorities. Consequently, the production methods must still be flexible enough to deal with many different specifications.

As with the all-metal bodies, as much standardization as possible has been introduced into the design of the composite bodies. This allows the maximum effective use to be made of the machinery in the well-equipped sawmill. There is also a resemblance between the work in the all-metal body and the composite body departments in that great use is made of adjustable jigs in the production of sub-assemblies for composite cabs. One of these jigs is shown in Fig. 8.

The jig is completely fabricated from timber and any necessary adjustment can be made quickly by unscrewing parts that act as datum points and then refixing them in the desired position. Even the clamping cams are made of wood. To ensure that the cab as assembled from the various sub-assemblies is correct, every cab is checked against a static fixture immediately after assembly. This fixture is shown in Fig. 9.

As far as possible all the sub-assemblies for composite body frames are jig built. Nevertheless, there is generally considerable work to be carried out after the frame has been mounted on its chassis. Since the type of work differs considerably as between frames for bus bodies and frames for fire engines, the key operatives deal only with one type of vehicle. It is found that this improves output rates. The framing for a commercial passenger vehicle is shown in Fig. 10.

There is little to be said about the remaining departments, except that they are organized to deal with small quantity production in an efficient and expeditious manner. The general procedure in these departments follows conventional lines for high quality work. Typical Dennis commercial passenger and fire-fighting vehicles are illustrated in Figs. 11 and 12.

AN OVERDRIVE UNIT

A Detailed Description of the Laycock De Normanville Design

WHERE the transmission of a car involves the use of a step-by-step gear box and the operation of a clutch pedal and a gear lever with some judgment for each change of gear, it is clear that, for the great majority of the public, a moderately low top gear ratio is desirable. For those enthusiasts who take pleasure in the skilled handling of the car, there have always been available, on the English and Continental markets, a limited number of vehicles with four-speed boxes, a very high top gear and a close-ratio third speed.

The general demand, however, is for rapid acceleration and effortless driving, and this has tended to produce vehicles which, at sustained high speeds, are running beyond the peak of the engine power curve. In this region, valve-bounce is imminent and the specific consumption curve rises steeply. Especially on the high-speed roads common on the Continent and in America such cars use more fuel than they should do, are more "fussy" in the engine and are probably wearing it out faster than need be.

This is particularly true of American cars, which are so commonly equipped with three-speed gear boxes in place of the four speeds generally used in this country and on the Continent. It is, therefore, not surprising that American designs embody some form of semi-automatic "overdrive" mechanism. Accepted practice in this respect involves the use of a simple epicyclic train giving an overdrive ratio of about 0·7 to 1, this being, incidentally the closest ratio conveniently possible in the smallest possible space. When fitting this mechanism it is common to lower the back axle ratio by about 10 per cent so that, compared with the standard vehicle the overdrive is actually of the order of 0·8 to 1.

Control is semi-automatic. Starting in direct drive, the power being transmitted by a free-wheel, a centrifugal governor operates a switch, which, on release of the accelerator pedal at speeds above about 30 m.p.h. energizes a solenoid engaging a locking pawl on the sun-wheel of the epicyclic gear as soon as synchronism is reached by the slowing down of the engine. On depressing the accelerator again the vehicle proceeds on overdrive at all speeds as long as the engine is driving.

On slowing down below 30 m.p.h. and releasing the pedal the pawl is withdrawn and the car reverts to direct drive. By depressing the accelerator to the maximum the driver can cut the ignition circuit for an instant and make the mechanism change down at *any* speed. Thus direct drive can be used for acceleration up to say 60 m.p.h. if desired.

It should be noted that the scheme provides a free-wheel on direct drive but a solid transmission on overdrive. The engine is thus normally available as a brake on overdrive but not on direct, or lower, gears unless a separate locking lever is operated which cuts out the overdrive altogether. Another point to be noted is that the engine must be allowed to slow down to synchronous speed, or below it, before the overdrive will engage. It is not, therefore, possible to make a quick change up, using the kinetic energy of the flywheel to boost the operation.

The De Normanville unit

The overdrive developed by Auto Transmissions, Ltd., and made by the Laycock Engineering Co. differs in many respects from the American system. In the first place, a compound epicyclic train is normally offered, so that an overdrive ratio of say 0·82 can be provided. The unit is therefore suitable for fitting as an optional extra to a standard chassis without altering the back axle ratio or giving an excessively high overdrive ratio. No automatic change is embodied, control being by a finger-tip lever on the steering column, which can conveniently act as the knob of the normal change-speed lever also, this being lifted by the finger, after changing into top gear, when overdrive is desired. Engagement is by oil-wetted cone clutches operated by hydraulic pressure, a free-wheel being provided in the direct drive so that the engine cannot "race" when changing down. The cone clutch locking the epicyclic train in the direct drive position is arranged to come into action fairly slowly, avoiding any "back snatch" through engagement before the engine has speeded up. When engaged it gives a solid drive and thus the engine is at all times available as a brake.

Changing into overdrive by a flick of the lever operating a small control valve, causes the locking cone to be withdrawn and the cone forming the brake for the overdrive epicyclic sun-wheel is engaged. The accelerator pedal need not be released during this operation and full power is transmitted by the free-wheel until the brake is sufficiently engaged to "take over" on the overdrive. Full use can be made of the kinetic energy of the flywheel in changing up, and this, coupled with the fact that full throttle is used throughout, is of considerable assistance when changing under adverse conditions.

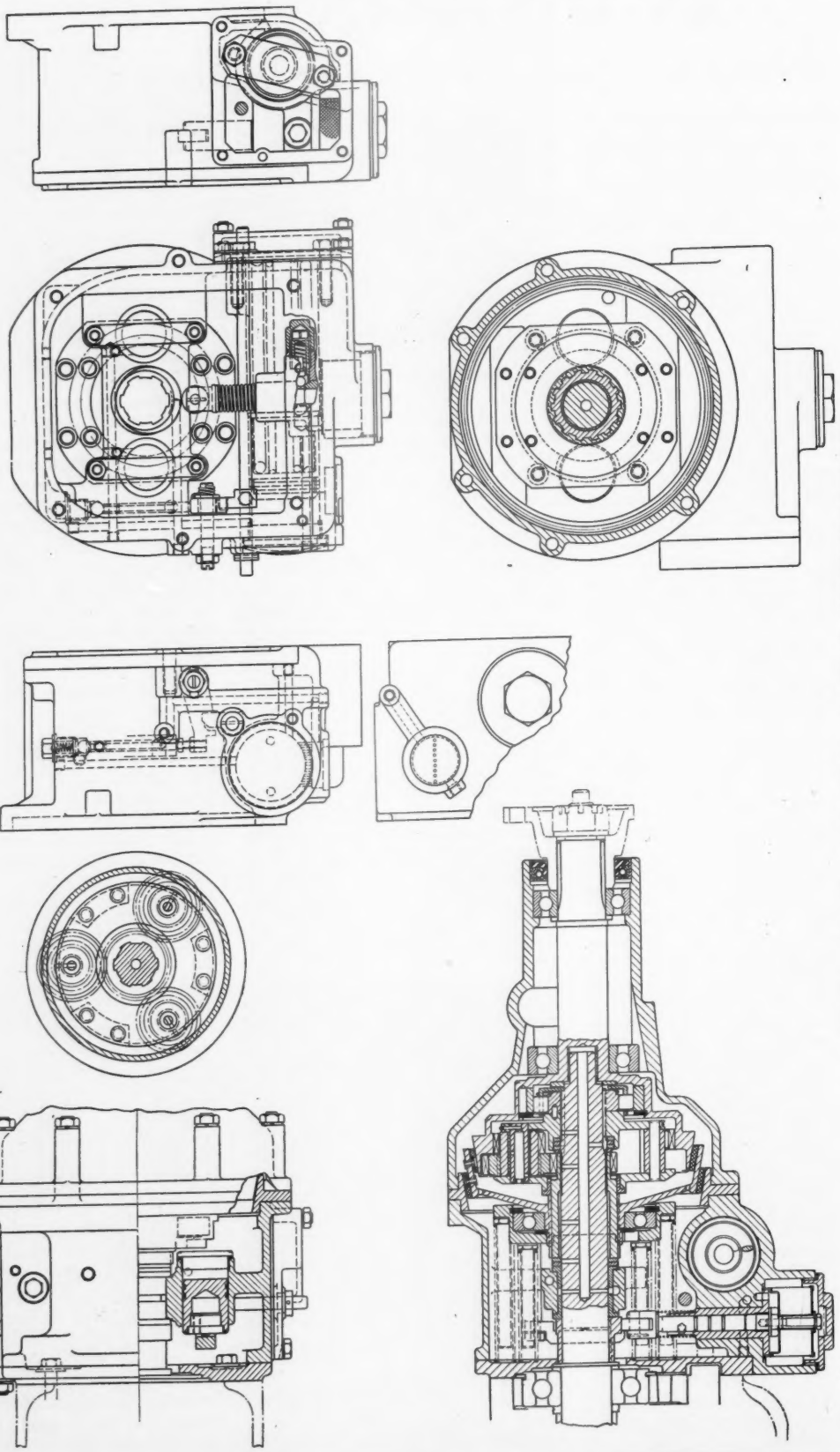
It will, of course, be understood that the clutch pedal is not operated when changing the overdrive gear. By combining the operating lever with the change-speed lever it is arranged that direct drive is engaged by the downward movement of the knob before the lever can be moved to change down to a lower gear in the synchromesh gear box. In other words, the makers intend the mechanism to be used as an overdrive only and not to give intermediate ratios on the lower gears.

Before describing the mechanism in detail it may be mentioned that the makers are offering, for commercial vehicle use, a somewhat similar unit but arranged to come in front of the gear box and to give an "underdrive" of about 1·3 to 1.

This is controlled in the same manner but is intended to be used with a slightly higher axle ratio and to give a double range of ratios, direct or underdrive being usable on all or any of the gear box ratios. Apart from the high gear available when running light or under favourable conditions, the frictional engagement of the De Normanville mechanism permits a change up from underdrive to direct to be made under load, without releasing the accelerator. This possibility should be valuable on a lorry in a hilly district, enabling a change up to be made under somewhat adverse conditions impossible with a dog-clutched two-speed axle.

The overdrive mechanism

Housed in an aluminium casing located behind the gear box, the De Normanville unit can conveniently take the place of the rearward extension of the gear box now common as a means of shortening the propeller shaft. The driven shaft is extended to carry first a cam operating the oil pump, and then a steady bearing with two opposed plain bushes providing



GENERAL ARRANGEMENT OF THE LAYCOCK DE NORMANVILLE OVERDRIVE

oil feed to the running gear. Next, is the sun-wheel of the epicyclic gear, this having 21 teeth and being carried on a Clevite bush. Beyond this the end of the shaft is splined to take the planet carrier and the hub of the direct-drive free-wheel clutch. The reduced end of the shaft is steadied by a plain bush carried by the final output shaft, which is supported in the unit rear housing by two ball bearings.

The illustration shows the annulus of the epicyclic, with its external coned surface for the locking clutch, made integral with the output shaft. In production, these will be made in two parts, with a flange connection.

The outer ring of the freewheel clutch is pressed and dowelled into the annulus member, the clutch itself is of the caged type, loaded by a clock-type spring made of round wire.

Three planet pinions are used; each one consists of a 15-toothed pinion with part of its teeth reduced to form shallow splines, and on to these a 24-toothed wheel is pressed, meshing with the sun-wheel.

The ratio given is 0·82 to 1, or $\frac{1 \cdot 22}{1}$ overdrive when the sun-wheel is held stationary, but, on account of the compound planet pinions it is quite easy to provide any ratio that may be suitable for any particular car. The sun-wheel and all planet pinions are made of EN 24 steel, cyanide hardened to a case depth of about 6/1,000in and tempered at 200 deg Centigrade, after gearcutting. The annulus is also made in EN 24 steel but merely heat-treated to 50/55 tons tensile before machining.

The annulus and the smaller planet pinions have teeth cut to a helix angle of 12 deg 20 min, while the sun wheel and the large planet pinions have teeth with a helix angle of 23 deg. Thus, in spite of the difference in tooth loading on account of the different diameters, an almost complete balance of end thrust is attained, amounting to only 8lb on each pinion. End thrust on the sun-wheel, amounting to about 100lb is taken by alternate bronze and steel thrust washers, lubricated by radial holes in the gear box driven shaft. It should be noted that these washers are stationary when in overdrive and only momentarily take a running load, during the period of clutch slip when engaging overdrive. The pinions have Clevite bushes and run on case-hardened pins, the load on each being only 276lb, and the relative speed of the pinion on the pin 875 r.p.m. at 1,000 r.p.m., and 1,280 lb in engine torque.

Splined to a forward extension of the sun-wheel is a cone member lined both inside and out with Mintex H 2 material at an angle of 20 deg inclusive. The inner lining engages the outside of the annulus stamping to give direct drive, while the outer lining works in a cast-iron ring sandwiched between the front and rear parts of the unit housing.

To the hub of the cone member is secured, by a Seeger circlip, a ball bearing housed in a flanged ring. This ring carries on its forward face eight pegs acting as guides to eight compression springs by which the ring, and with it the cone member, are thrust against the annulus cone, locking the direct drive.

Also screwed to the front face of the ring are four studs picking up two crossheads against the rearward faces of which bear two pistons working in cylinders formed integral with the front part of the unit housing. The cylinders are connected by drillings to a vertical bore in which slides a hollow spindle with a reduced upper end. An enlargement of the top of the bore contains a ball seating downwards on a shoulder, and oil is supplied under pressure to the chamber containing the ball, which is thereby held on its seat.

A cross-shaft connected to the steering-column control operates a finger which lifts the hollow spindle when overdrive is required. The movement of the spindle first seats its reduced end against the ball, which is then lifted, admitting pressure oil to the space round the reduced end of the spindle, whence it passes by the drillings to the cylinders. The pistons then overcome the springs and move the cone member forward until the holding clutch is engaged, the sun-wheel brought to rest, and overdrive given is in operation, the freewheel over-running.

Rotation of the cross-shaft in the opposite direction lowers the hollow spindle, allowing the ball to come on its seating and cutting off the oil supply to the cylinders. Further movement brings the spindle out of contact with the ball, allowing the oil to escape down the hollow spindle and the cone member to move towards the direct drive position.

The passages are deliberately restricted so that full engagement of the direct drive clutch takes about half a second. This gives the engine ample time to pick up speed and take up the drive on the free-wheel before the locking clutch engages. This is a valuable feature, as the clutch must be loaded sufficiently to take the torque set up by the reverse gear and this is about three times that required for direct drive. Sudden engagement of the clutch would therefore cause an undesirable "back-snatch."

It should be noted that the actual resultant torque handled by the clutches is very much less than the final output, less than 20 per cent in fact, since the ratio of the epicyclic is not far removed from unity. The oil is supplied by a pump, worked by a cam on the gear box driven shaft, having a diameter of $\frac{3}{8}$in and a stroke of 5/32in. The pump barrel is pressed into the front housing and the delivery ball valve seats in a counterbore in a cross-drilled hole in the barrel. Suction is by a port and the quantity is about 0·01 cu in per stroke.

From the pump the oil passes to a horizontal accumulator cylinder, in which a piston moves against a preloaded compression spring. Normal pressure is about 460lb per square inch on reaching this the piston uncovers relief holes, the oil from which is led by drilled passages to the annular groove between the two steady bushes on the gear box driven shaft. Radial holes in the shaft feed, by an axial drilling and other radial holes, the sun-wheel bush and the inside of the planet carrier with its planet pins.

The quantity of oil supplied, while quite adequate, is not sufficient to cause undue loss by churning inside the annulus. No reliance is placed, as in some overdrive mechanisms, on unassisted splash feed liable to be affected by the oil level.

A fine filter, supplemented by a magnet, is provided on the pump suction. Both accumulator and clutch-operating pistons have special cast-iron piston-rings, consisting of an "L" shaped ring with an additional narrow ring in the groove of the L, spanning the gap in the former. These rings have been known to hold oil pressure for several days, though it will be understood that this is of no importance in the operation of the mechanism. When starting from rest the direct drive is used (engaged by the freewheel and springs) and long before the overdrive is required the pump will have built up sufficient pressure. Direct drive is also always available, for example when towing to start the engine, etc.

During a short run in a Standard "Vanguard" fitted with this overdrive, the ease of control was at once appreciated and even on what is normally a quiet car the reduction in noise at high speeds appeared to be of substantial benefit. Absolute control over the use of the overdrive is undoubtedly a valuable feature.

THE HYDRAULIC SERVO SYSTEM

Details of the Lockheed Design as Fitted to the Daimler CD650 Chassis

REDUCTION of the physical effort that has to be exerted by the driver in applying the brakes of a heavy vehicle is the main objective in the developments of air, vacuum and hydraulic systems of assistance, and in the various combinations of these. Regular increases in the size and weight of vehicles, the requirement of the carriage of heavier loads at higher speeds and the fitting of larger section heavier tyres are all factors that have made the problem progressively more important. This is especially so in the case of public service vehicles that must also meet the added burden of halting at closely spaced stopping places—while adhering to a running schedule that leaves no time to spare.

There has also been a demand for improvement in overall efficiency of vehicles, and for the provision of greater stopping power from existing sized brakes. This is due to the necessity for avoiding radical redesign and also the difficulty of accommodating larger components on chassis necessarily restricted in size by regulations and other reasons.

Largely for the same reasons the necessity for reducing steering effort is making itself apparent. While this aspect of the problem has been under consideration for many years, little progress has been made until recently. At the last Commercial Vehicle Exhibition at Earls Court there was seen for the first time a British vehicle fitted with a complete hydraulic servo system, designed not only to operate the brakes, but also the steering and to aid actuation of the gearbox, the handbrake and other controls. This vehicle is the Daimler CD.650 chassis, described in the *Automobile Engineer* for June, 1949, designed to carry a 56-seat double-deck omnibus body with an all-up weight of 12 tons and which is integrally fitted with the Lockheed Hydraulic Servo Accumulator system.

Before examining the system in detail it is necessary to appreciate the underlying principles of hydraulic assistance and the varying manner in which it is applied.

In a hydraulic braking system the aim is to transmit power with the minimum loss of mechanical power. The Lockheed organization claim 90 per cent efficiency in the transmission of braking effort. The amount of assistance desirable, in order to obtain adequate braking power under all conditions with a heavy vehicle is said by Lockheed to be a boost of between 3 and 5 to 1 between pedal and brake shoe.

It is reasonably claimed that a hydraulic brake in proper working order is both smooth and progressive in action and that the driver can feel exactly what effort is being made. He is thus in a position to reduce the speed of the vehicle in a sensitive manner or to make an emergency stop as required. In each case the amount of effort required is proportional and should always be the same for an equal result. The addition of a servo should ideally reduce the foot and leg effort required but still leave the brakes fully responsive so that the driver can "feel" them in use. Servo systems should not, in any circumstance, dominate the driver or anticipate his actions.

Towards the fulfilment of these aims the Automotive Products Company has progressively developed the Lockheed fluid braking system from the simple use of a hydraulic medium to operate the braking system of a private car, to the latest hydraulic servo accumulator system. This not only operates the foot brake, but will also assist in the operation of all the principal controls of the heaviest type of commercial vehicle chassis including the steering.

The several units comprising the hydraulic servo accumulator system and their method of operation are as follows:—

The Pump. The heart of the system is a seven-cylinder radial, plunger type, pressure pump. The mark VI pump is quite a small unit measuring some 5in diameter and 4.5in wide, and is designed to produce high volumetric efficiency over the whole speed range. Basically it consists of a two-piece outer cover, one part known as the fixing flange, the other as the high pressure end cover. To this end cover are fitted the seven separate nickel steel cylinders. They are positively located and bed down on high pressure rubber seals, being secured by two studs with nuts locked by a tab washer.

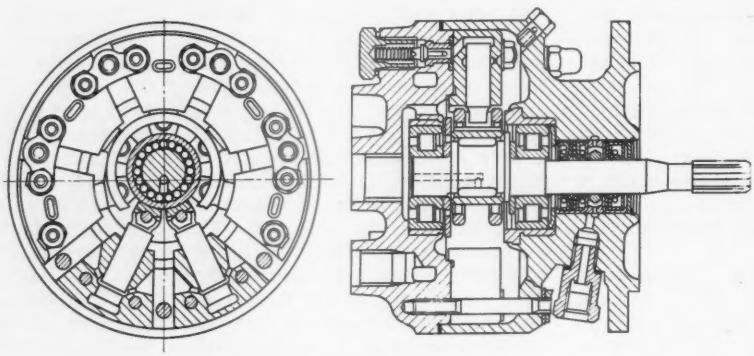


Fig. 1. Engine-driven fluid pump.

Each cylinder is bored to take a plunger, the bore being lap finished. Towards the closed end of the bore are two grooves, the bottom one of which connects with three inlet ports and the top groove with a single outlet port. This outlet port, normally closed by a spring-loaded non-return valve, communicates with a cored passage in the high pressure end cover, the passage leading to the pump outlet.

The pump plungers are machined from tool steel, that surface which forms the piston also having a lapped finish. Each plunger is operated through a shoe which is in contact with the bearing ring around the eccentric, the shoe being grooved lengthwise to receive a pin which passes through the eye of the plunger. The eye does not completely surround the pin, so that, while the plunger cannot disengage itself, the load pressure is transmitted through the pin to the shoe without subjecting the pin to shear stresses.

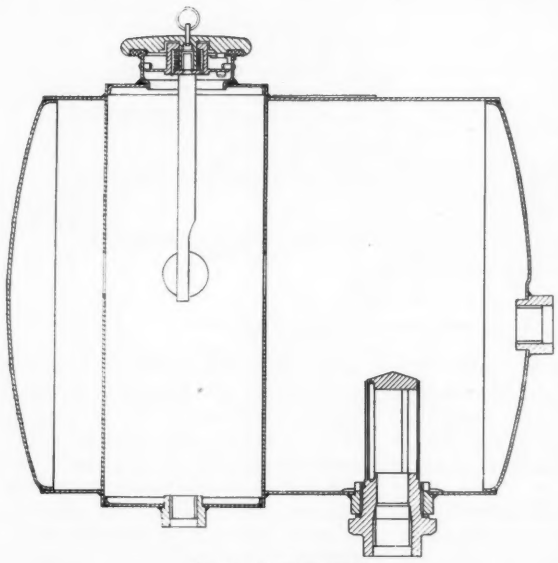


Fig. 2. Header tank.

The ends of the pin are cut away to approximately half their diameter and two shoe retaining rings hold all the shoes and pins on to the bearing ring and ensure positive retraction of the plungers.

Carried in two roller bearings that are housed in steel liners in the end covers, the nickel steel forged pump shaft has an integrally formed eccentric. Surrounding the eccentric are 21 needle rollers and outside them is the plain bearing ring referred to above. The inner end of the shaft is drilled axially to meet a passage drilled diametrically into the eccentric to allow fluid to pass to the needle rollers for lubrication purposes. The drive end of the shaft terminates in a splined portion to take the drive. It is drilled internally for lightness.

The fixing flange and high pressure end cover are fastened together by seven studs and cap nuts which bed down on to copper washers. Three Gits seals surround the driving end of the pump shaft to prevent fluid leakage. The inner two are back to back and are separated by a spacer from the outer seal, outside which is a dust cover. The whole seal assembly is retained by a Seeger circlip. Lubrication of the pump is effected by the fluid circulating within it.

The pump takes some 2 b.h.p. when pumping at 1,000lb sq in, turning at 2,000 r.p.m. When no power is being taken from it little driving effort is required. The pump is normally arranged to rotate at 3,000 r.p.m. for a vehicle speed of 30 m.p.h. Operation is equally effective in either direction of rotation, so that the drive can be taken from any convenient point on the engine, the gearbox or by vee belt from a pulley on the propellor shaft. There are only two connections. An inlet for fluid from the supply tank is centrally positioned in the high pressure end cover, the outlet being on the same face.

The method of operation of the pump is as follows: As may be seen from the section illustration of the pump (Fig. 1), there is a considerable cavity in the centre of the pump and this is flooded with fluid. The eccentric draws down each plunger in turn so that the groove in the cylinder in which it works is uncovered to the fluid in the body of the pump. As each plunger moves up its cylinder, it covers the inlet port trapping some fluid in the cylinder, up which it is forced, thence past the non-return valve into the annular passage cast into the high pressure end cover, thence to the pump outlet. The inclusion of an accumulator in the system means that operation of the servos does not depend upon the pump being running.

Supply Tank. This is a two-piece unit of welded construction, the main tank being a horizontal cylinder 7·75in diam, 10·25in long. Towards one end there is a vertical cylinder having an internal diameter of some three inches. Intercommunication between the two compartments is provided by four 0·875in holes slightly above the centre line of the horizontal part of the tank. The large diameter spring loaded filler cap is over the vertical portion of the tank. It is fitted with a dip stick.

The fluid supply to the pump is taken from the bottom of the main tank through an internal filter, the supply to the master cylinder being taken from the bottom of the secondary tank. Surplus fluid is returned to the tank via a banjo connection at one end of the main tank. The tank is usually fitted on the dash under the bonnet.

Cut-out Valve. A small spring-loaded valve, with an adjustable setting, which controls the supply of fluid from the pump according to the amount of pressure in the accumulator. When the accumulator is empty, or until such time as the fluid pressure within it reaches the predetermined upper limit, usually about 1,200 lb sq in, the fluid enters through the inlet E, as shown in figure 3. Being unable to pass the valve seat J, as the valve C is held up to it by the combined power of the springs K and L, it unseats the sleeve valve D and passes out through the outlet F to the accumulator. The fluid, under pressure from the pump, continues to charge the accumulator until such time as the rising pressure exerts a greater force on the valve at area A than the opposing force of the two springs K and L. The valve is gradually unseated and fluid passing through presses on the flutter plate H which thereby compresses the spring L, thus relieving the combined spring load on the valve C of the effect of L.

The pump flow having then only to move the valve C against the one spring K, is able to move C still further back until a free escape of fluid through G is effected. With this phase there is then a complete drop in pressure through E and G and the valve D is forced on to the needle B by the high accumulator pressure and the spring M. The full accumulator pressure acting on area N is then applied to the seat at B, positively

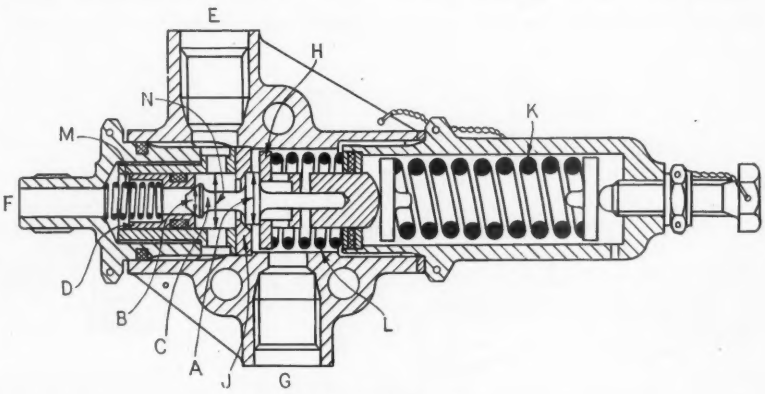


Fig. 3. Cut-out valve.

THE HYDRAULIC SERVO SYSTEM (Continued)

forcing the needle valve C back to the close coiling of the spring L, in which position the pump flow can freely circulate through G and back to the supply tank.

When the pressure in the accumulator has fallen to the point where the combined load of the two springs is greater than the effect of the accumulator pressure acting on area B, then the valve C is moved back until the knife edge A reseats itself on J. The pump flow being again checked through J, repeats its original operation by unseating D and recharging the accumulator through the outlet F.

Accumulator. The latest type of accumulator consists of a solid drawn steel cylinder of great strength, 3in diameter by 18in. in length. Within the cylinder, and secured at one end, is a reinforced rubber bag. The end at which the bag is secured is known as the air end, the other being the fluid end. The air side is inflated, through a special Schrader connection, by external means to an initial pressure of some 450-500 lb sq in. The free air capacity of the accumulator is 100 cu in. The fluid side is connected through a banjo connector to the cut-out valve, via the filter. (See Fig. 4.)

Fig. 4. Bag type accumulator.

Fluid entering the accumulator from the pump, compresses the rubber bag and is immediately itself placed under pressure within the accumulator by the air pressure in the bag. An effective working pressure is therefore created as soon as delivery commences from the pump. The energy stored in the accumulator is sufficient to enable a large number of brake, gear, or other applications to be made even should the servo pump not be working or fail in service. The small overall size of the accumulator unit renders it a simple matter to accommodate one or two within a chassis frame side member or in other suitable place.

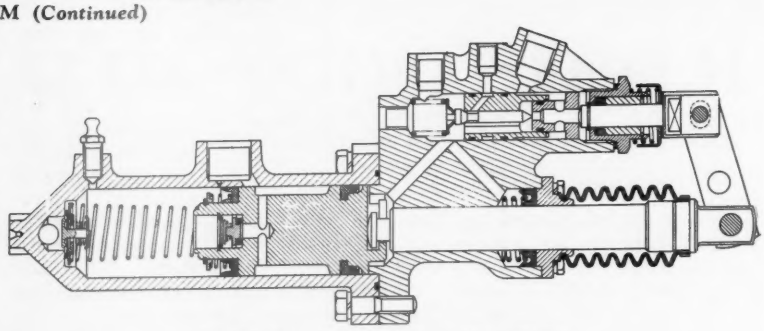


Fig. 5. Control valve and master cylinder.

Control Valve and Master Cylinder. Built in one unit this combines the normal master cylinder with the valve which controls the servo action, accepting pressure energy direct from the pump, or, when necessary from the accumulators. The control valve, as will be seen from the section illustration (Fig. 5), is mounted in a separate bore above that of the master cylinder, the two pistons being connected together, and in turn to the brake pedal, by a balance lever,

Three separate functions are combined in this unit:—

(a) The normal master cylinder, which is fed with fluid direct from the inner (vertical) compartment of the supply tank, directly operates the rear brakes only.

(b) The servo chamber, behind the main piston of the master cylinder, in which the fluid from the pump (or from the accumulator under certain conditions) acting on the back of the piston aids the application of the brakes.

(c) The control valve which regulates the entry of fluid under pressure either from the pump or from the accumulator also operating the front brakes by direct servo pump pressure. There are therefore two independent methods of applying front and rear brakes through pressure on the one pedal. The pedal pressure is quite light and is variable as required. The maker's recommendation is 100lb for 16ft seconds.

Operation of the brake pedal first closes the release valve thus restricting passage of fluid through the return port to the supply tank. Pressure is thereupon built up behind the main piston, the effective area of which is approximately three times that of the release valve. Thus a boost ratio is formed which is effective at all pedal pressures. Should insufficient pressure be delivered by the pump, as for example, should the engine be stopped, further application of the foot pedal will push a balance valve off its seat and uncover the accumulator port thereby permitting pressure to enter from the accumulator and so maintain servo pressure. The return of the foot pedal opens the release valve restoring free circulation.

Solid drawn copper pipe is used to connect the several units except where flexibility is required. As the pump is generally fitted to either engine or gearbox, provision has to be made for movement due to flexible mounting. A length of Lockheed high pressure flexible hose is therefore fitted between the pump outlet and the supply line. Similarly, to allow for movement of the dash, on which the supply tank is commonly located, all the connections to and from the tank are fitted with a short length of Avery low pressure flexible hose.

The five principal units, with a junction box and the necessary connecting tubing, form the essential basis of the accumulator system which can be built in to almost any type of chassis. The system can, of course, be extended by adding auxiliary servo units for the operation of clutch pedal, gear engagement on a self-change gearbox, or handbrake. The power operation of doors is also

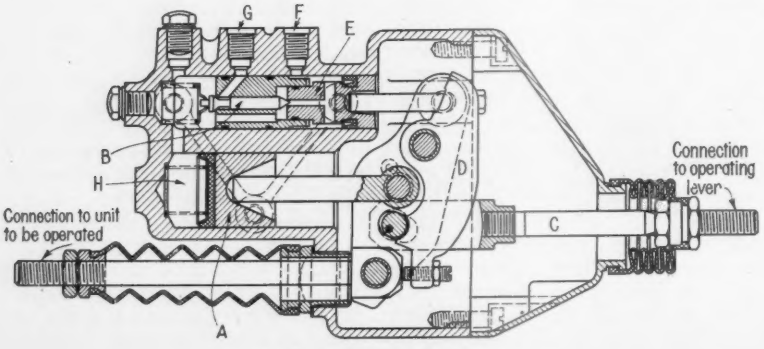


Fig. 6. Hydro-mechanical servo.

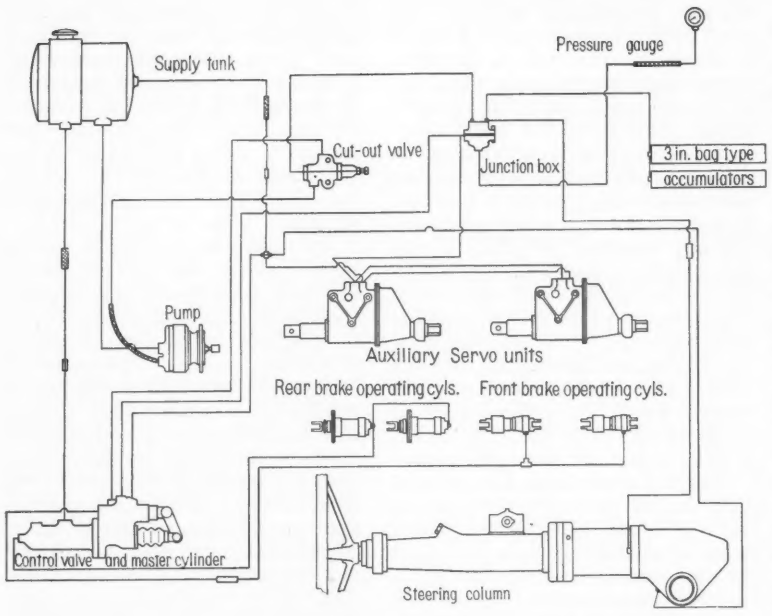


Fig. 7. Hydraulic layout for Daimler CD.650 chassis.

within the capacity of the equipment.

Auxiliary Servo Units measure 16in long by 6·5in high by 3·25in wide and weigh 13lb. They are designed to give a boost ratio of 5 to 1 and can produce a maximum effort of 2,840 inch lb at 1,000 lb sq in accumulator pressure. It is important to remember that should, by mischance, the accumulator pressure fail the direct mechanical effort remains. In the case of an assisted handbrake, therefore, its efficiency does not depend upon the amount of fluid pressure available at any particular moment. Internally the unit comprises a 1·5in diameter hydraulic ram, a reducing valve and a compensating operating link to which is connected the ram piston rod, the valve piston rod and both input and output operating rods. (See Fig. 6.)

Fluid pressure is applied to the hydraulic ram A having been admitted by the valve B, control of which is synchronized with the mechanical load delivered. When the handbrake lever, for example, is operated it pulls forward the rod C and the operating link D applies an equal reaction on to the reducing valve. This pressure applied to the link, first to the release valve E closes the gap between the release valve and the reducing valve, shutting off the connection F to the supply tank and opens the reducing valve, admitting high pressure fluid from the accumulator, via port G, to the pressure annulus H of the ram. This working pressure will build up until its reaction, acting over the diameter of the reducing valve, equals the load placed upon the valve by the link. When this occurs, the reducing valve will re-seat itself and shut off further accumulator pressure.

A further pull on the handbrake lever would open the valve still more, and admit a higher pressure until this greater load was equalled, and a reduction in the handbrake pull would allow the trapped ram pressure to force open the gap between the release valve and reducing valve, permitting pressure to escape until this new out-of-balance force had been opposed.

Installation on the Daimler CD.650. As previously stated, this chassis, designed to carry a 56-seat bus body and permitting an all-up weight of 12 tons, is the first British vehicle to be equipped throughout with a complete hydraulic system operating or assisting all major controls. The layout diagram of the several components and the method of inter-connection is shown in figure 7.

Steering Unit. It has been found possible to incorporate the power assistance within the steering column while retaining normal steering. The standard type worm and nut have been lifted towards the top of the steering column to provide space for the hydraulic cylinder. A ram, within the cylinder, is connected by a rod to the steering nut above it and by a link to the steering shaft operating link below it. This direct mechanical connection ensures that manual operation is at all times available should a fault develop in the hydraulic system. Care has been taken to maintain the usual degree of self-centring action.

Reference to figure 8 will show the

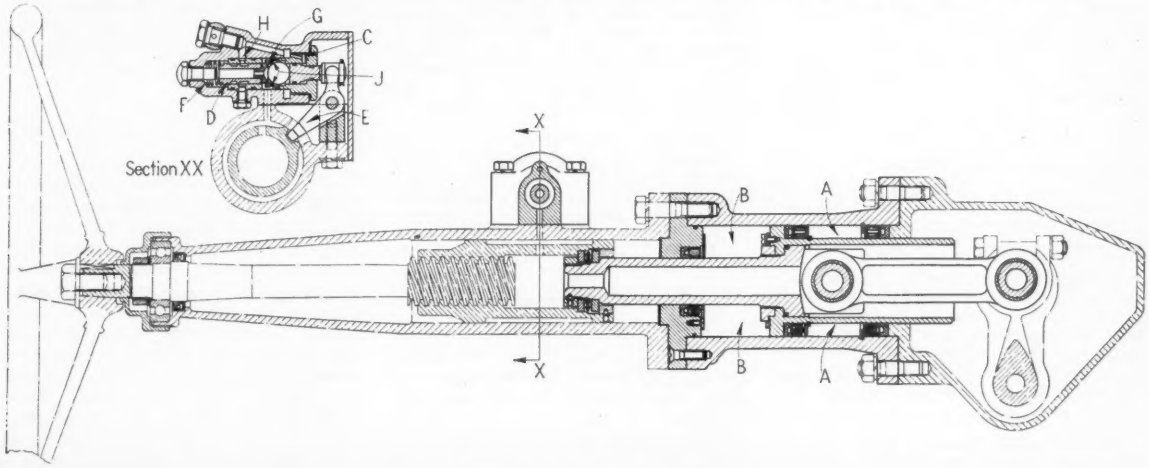


Fig. 8. Assembly of servo steering gear.

THE HYDRAULIC SERVO SYSTEM (Continued)

construction. The ram is double acting but the effective areas on the two sides are in the ratio of 2 to 1, the larger being on the upper side at B. Pressure is admitted to the smaller side at A direct from the accumulator. Pressure on the upper side is controlled by a valve which normally allows a pressure equal to half the accumulator pressure to enter chamber B. It will be seen that with full accumulator pressure on the small area opposing half accumulator pressure on the large area the gear is in balance. Steering is effected by raising or lowering the pressure in B above or below the mean value of half accumulator pressure so that the pressure in B overcomes or is overcome by the constant accumulator pressure in A.

The control valve is positioned halfway up the steering column at point X—X on the drawing in figure 8, in which the valve is shown in section. The sequence of operation is that within the valve, fluid from the accumulator enters through the port H, feeding the balanced valve D which is normally closed by a light spring. The accumulator pressure at H is also fed into the annular space C, the effect of this being to apply a thrust to the ball. This ball is therefore forced into contact with the valve seat at G and unbalancing the valve D admits fluid from the accumulator into the space F which is in communication with the area B in the steering column, allowing pressure to build up until such time as it exerts a force on the ball equalling the effect of the accumulator pressure in annulus C. The reducing valve will then close. The pressure acting on seat area J is then balancing the accumulator pressure acting on annulus C. As the ratio of areas J to C is 2 to 1, the pressures in F and H, also B and A, must be in the ratio of 1 to 2. As area B is twice area A the ram is in balance.

The bell crank lever E is engaged in a slot of gear tooth form in the steering worm, therefore the torque reaction transmitted from the worm to the nut exerts a force in a clockwise or anti-clockwise direction on the bell crank lever during steering. This force adds to, or reduces, the hydraulic load constantly applied to the ball. The result is to increase the pressure in annulus F and chamber B until the increased load has been balanced, or to allow an escape of some of the fluid until the pressure in F and B has dropped sufficiently for the net force on the ball to re-seat the ball at G. Fluid escaping past the seat G passes through ports into the centre of the steering column and drains into a level common to the header tank.

The pressure in area B during steering either rises or falls, forcing the ram downwards or upwards according to the direction and degree of unbalance in chambers A and B. As the rise and fall of pressure in B is always proportional to the pull on the steering wheel the servo power is also proportional to the effort of the driver. On release of the steering torque at the wheel the pressures will readjust themselves as the reducing valve assumes equilibrium. This readjustment permits the normal self-centring action to the steering wheel. The whole movement is practically instantaneous and there is no apparent time lag in operation. Lubrication of the moving parts of the steering column is effected by the fluid within it.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

The following meetings will be held during January:—

BIRMINGHAM CENTRE

Tuesday, 24th January, 6.45 p.m.— General Meeting in the James Watt Memorial Institute, Great Charles Street. Paper: "Internal Expanding Shoe Brakes for Road Vehicles," by Ivan Waller, B.A., A.M.I.Mech.E.

COVENTRY CENTRE. *Graduates' Section.*

Tuesday, 17th January, 7.15 p.m., at the Hare and Squirrel Hotel. Brains Trust by members of the senior Centre Committee.

LUTON CENTRE

Tuesday, 24th January, 7.15 p.m.— General meeting in the Assembly Room, Town Hall. Paper: "Fuel Anti-knock Requirements for Automobile Engines," by C. G. Williams, D.Sc., M.I.Mech.E.

NORTH-EASTERN CENTRE

Wednesday, 18th January, 7.30 p.m.— Address by the Chairman of the Centre, T. H. Parkinson, M.I.Mech.E.

NORTH-WESTERN CENTRE

Wednesday, 25th January, 7.15 p.m.— Annual dinner of the Centre at the Engineers' Club, Albert Square, Manchester, followed by the A.D. Chairman's Address, "Ignition Equipment—Development and Lessons," by E. A. Watson, O.B.E., D.Sc. (Member of Council).

SCOTTISH CENTRE

Monday, 16th January, 7.30 p.m.— General meeting at the Institution of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper: "Fuel Anti-knock Requirements for Automobile Engines," by C. G. Williams, D.Sc., M.I.Mech.E.

WESTERN CENTRE

Thursday, 26th January, 6.45 p.m.— General meeting at the Royal Hotel, Bristol. Paper: "Piston Assemblies for Road Transport Oil Engines," by J. L. Hepworth, B.Sc.(Eng.), A.M.I.Mech.E.

The following meetings will be held during February:—

LONDON

Tuesday, 14th February, 5.30 p.m.— General Meeting at Storey's Gate, St. James's Park, S.W.1. Paper: "Some Factors Governing the Performance of Crankcase Lubricating Oils," by A. Towle, M.Sc.(Eng.), M.I.Mech.E.

COVENTRY CENTRE

Tuesday, 7th February, 7 p.m.— General Meeting in the Geisha Café, Hertford Street. Paper: "Internal Expanding Shoe Brakes for Road Vehicles," by Ivan Waller, B.A., A.M.I.Mech.E.

TOWING TRAILERS

A PRIZE of 50 guineas is offered by the Caravan Club of Great Britain and Ireland to encourage and assist improvements in the road performance of caravans. It will be awarded for a paper on "The Dynamics of Towing Two - wheeled Trailers." It is felt that the road performance of caravans is capable of considerable improvement and that a wider and fuller grasp of the theory involved in towing would be of great assistance.

It is suggested that analysis of towing problems should result in useful design information concerning stiffness of frames, optimum axle setback, length of drawbar, optimum nose weight, choice of springs and tyres, and so on. It will also offer guidance to users on the question of towing brackets, weight distribution of the caravan contents, and the suitability of various types of car for towing purposes. The competition is open to anyone, and papers of from 3,000 to 10,000 words should be sent to the Secretary of the Caravan Club, 24, Store Street, London, W.C.1, not later than March 31st, 1950.

RUBBER IN ENGINEERING

THE London section of the Institution of the Rubber Industry are organizing a one-day Conference on the behaviour and testing of rubber under dynamic conditions. It will be held in London on March 17th, and arrangements have already been made for the presentation of six papers, which, together with others that may be arranged, will serve as a basis for a general discussion on the use of rubber in engineering applications. Those wishing to attend the Conference should write to the Institution at 12, Whitehall, London, S.W.1, as soon as possible. (1874)

Institution of Mechanical Engineers, Automobile Division

ROAD VEHICLE BRAKES

Characteristics of Internal Expanding Shoe Units

By I. M. Waller, B.A., A.M.I.Mech.E.*

The wealth of information already available on the subject of brakes is indicated by a prolific bibliography. There is little scope for new subject matter, the only novelty being in the approach to the problems. Emphasis is laid on the importance of discriminating between the characteristics of the linkage and of the brake units themselves, and the paper concentrates on the latter. Shoe characteristics are considered separately from those of the complete brake assembly, and a standard "three-to-one" shoe is evolved as a common basis on which to compare the different types of brake. Formulae are given for both shoe and brake factors and it is shown that, by using their reciprocals, the form of the equations becomes remarkably simple, and that, when thus plotted, the significant values are more easily read. The features considered most desirable in a brake are listed and a preference is shown for the two-leading-shoe brake using sliding shoes and hydraulic or pneumatic operation.

Some outstanding problems, such as improved cooling of the brake path, are raised. Wear, squeal, and fade are discussed, further developments in automatic adjustment are indicated, and a plea is made for improved oil and grease retention for hubs and axles to prevent contamination of brake linings. Brake testing is considered from first principles, and a routine for road testing is suggested. It is recommended that the investigation of temperature effects be confined to the laboratory.

The paper concludes with a reminder that it is not the characteristics of the lining as a whole, but those of the surface finishes prevailing at the moment, which are being observed, and that the basis of further progress is the study of surface phenomena and the search for better cooling.

IN any study of road vehicle brakes a clear distinction should be made between the brakes themselves and the mechanism which transmits the driver's effort on the brake pedal to each of the brakes on the vehicle. This discrimination is desirable because one make or type of brake is often used in conjunction with more than one make or type of linkage, and when power assistance is employed three different makers of proprietary components may contribute towards the braking system.

For example, when the Bendix Duo-servo type of brake is fitted on British vehicles it is usually operated by rods or cables, whereas on American cars this type is generally hydraulically operated. On a number of commercial vehicles, Girling two-leading-shoe brakes are operated by a Lockheed hydraulic system assisted by a Clayton-Dewandre vacuum servo device. On certain 'bus chassis are to be found brakes of the chassis manufacturer's own design operated on some models by the Westinghouse air pressure system and on others by the Dewandre vacuum servo, the same brake units being incorporated in each case.

There are many such combinations, but a detailed study of them is beyond the scope of this paper, which deals chiefly with the brake units themselves. In the notes on brake testing, however, the difficulties of distinguishing between the characteristics of the linkage and of the brakes themselves are emphasized.

Although disk brakes are growing in popularity, internal expanding-shoe brakes only are discussed.

Types of Brake Shoe. A significant trend in brake design is the increasing use of floating shoes instead of the conventional type with a fixed pivot. This development has been pioneered by the proprietary brake manufacturers, but has not yet been adopted for those heavy vehicles in Great Britain whose brakes are of the chassis manufacturer's own design It is therefore of current interest to review the comparative merits of the two classes of shoe, those with a fixed pivot being typified in Fig. 1 while types of floating shoe are shown in Figs. 2 and 3.

The fundamental difference between the two is that the floating shoe can centre itself in the drum when the brake is applied, whereas with a fixed pivot, concentricity is difficult to obtain and easily upset. On the other hand, when the brakes are released the fixed pivot provides a positive location, but the floating shoe requires some kind of frictional device to centre it in the off-position, and irregularities in the rotation of the drum have to be relied upon to knock the shoe clear should it rub at one end of the lining, while the friction device together with the pull-off springs hold it from floating back into contact with the drum under the influence of inertia forces.

Shoe Nomenclature and General Comparisons. There does not appear to be a generally recognized nomenclature to describe the two main classes of shoe and their subdivisions, although a defined classification was given by Acres (1946) :—

(1) Solid shoe with fixed anchor pin.
(2) Articulated or floating shoe.
(3) Pivoted shoe mounted on a separate carrier.

The first group might aptly be described as "pivoted"; a word which would then have to be avoided in relation to the floating types. "Non-floating" is not an attractive title, although adequate. "Anchored" is a possible alternative, but as ships float when at anchor it would be a misnomer if applied to shoes which do not float. "Rigid" has been used, but is unsuitable since the shoe may deliberately be given flexibility.

The following nomenclature is therefore proposed :—

(1) Pivoted shoes (Fig. 1).
(2) Floating shoes :
  (*a*) Sliding (Fig. 2).
  (*b*) Articulated (Fig. 3).

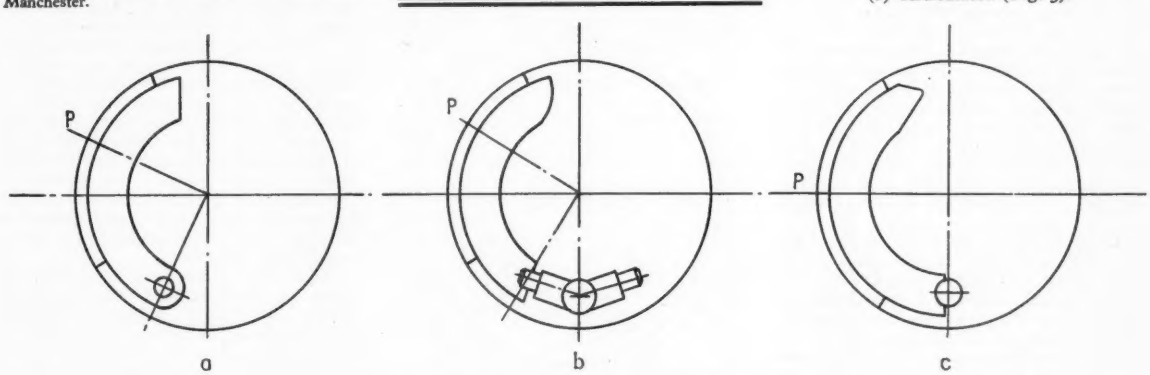


Fig. 1. Three Examples of Pivoted Shoes.

(*a*) Typical 'bus and truck shoe.
(*b*) Girling non-servo. P indicates position of maximum rate of wear.
(*c*) Girling hydraulic.

* Technical Manager, Small & Parkes, Ltd., Manchester.

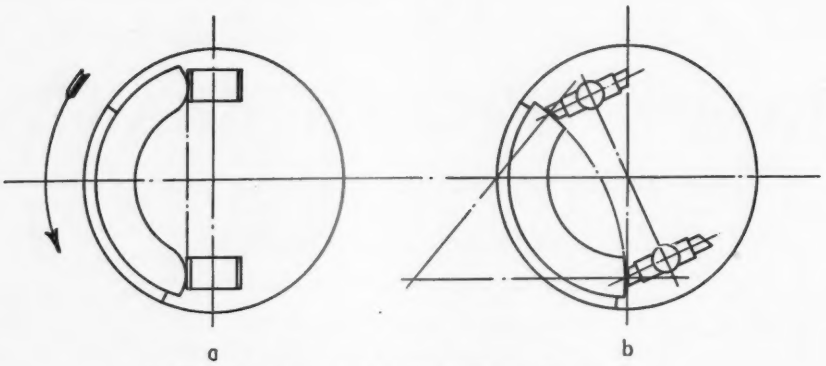


Fig. 2. Two Examples of the Sliding Type of Floating Shoe.
(*a*) Lockheed. (*b*) Girling two-leading shoe.

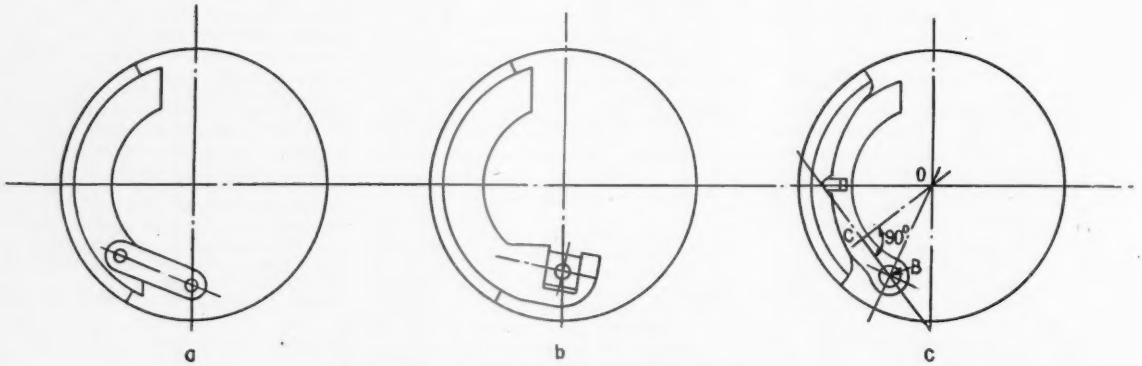


Fig. 3. Three Examples of Articulated Floating Shoes.
(*a*) Huck. (*b*) Lockheed slotted. (*c*) Lockheed tank.

Other items are defined as follows :—

Leading shoes : shoes in which the movement of the drum over the lining is towards the pivot or abutment.

Trailing shoes : shoes in which the movement of the drum over the lining is towards the applied load.

Toe and heel : the ends of the lining in relation to the " ankle " at the pivot or abutment.

Shoe tip : the part of the shoe at which the operating load is applied ; that is, the end remote from the pivot or abutment.

Pivoted Shoes. It is easy to predict, by the simple graphical method shown in Fig. 4, how the lining of a pivoted shoe will wear. With the shoe pivot as centre, an arc is struck passing through the centre of the brake. A circle of radius equal to that of the drum is then drawn from a centre on the arc at a distance from the drum centre equal to the maximum lining wear. The position of the point of maximum wear will lie on the line which passes through the centre of the circle and the drum centre, and which will make with the line through the pivot and the drum centre an angle of 90 deg. plus half the angular displacement of the shoe, which may be ignored.

The difference in wear between the ends and the middle of the lining can be reduced by making the shoe flexible, but it is desirable that the lining should be equally disposed about the point of maximum wear. Attempts to augment or reduce the effectiveness of a pivoted shoe by moving the lining towards the toe or the heel are likely to make it more sensitive to eccentricity or variations between the radii of the lining and of the drum. The possible effects of these three variables (position and eccentricity of lining and drum, and variation in lining and drum radius) on the bedding of the lining are shown in Table I.

The position in which the lining is mounted on the shoe varies considerably with different designs of brake. Both concentricity and the radius of the lining relative to that of the drum, apart from initial dimensional errors, can be upset by thermal expansion and mechanical deflexion during operation ; moreover, the general flexibility of the shoes, the carrier-plate assembly, the drums and the wheel bearings, all have to be taken into account when interpreting the bedding and the wear pattern.

Bedding at the toe of the leading shoe

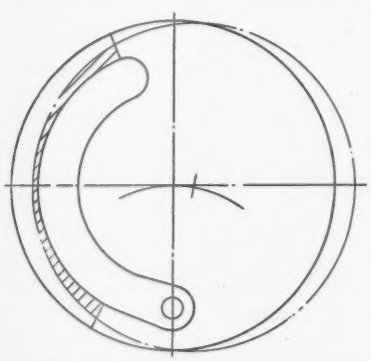


Fig. 4. Graphical Method of Predicting the Lining Wear with a Typical Pivoted Shoe.

will raise the power of the brake and in extreme cases lead to brake-grab, and this will be accentuated if the included angle of the lining is increased.

The usual method of obtaining concentricity is to jig-grind the linings to the desired radius after they have been fitted to the shoes and to rely on the relative accuracy of the jig and of the positioning of the pivot pins. American practice sometimes obviates errors by providing an eccentric adjustment for the pivots.

With the Girling N.S. brake, shown in Fig. 1*b*, the error can be divided between the two shoes by rotating the adjuster housing to its best position before tightening the setscrews securing it to the carrier plate ; moreover, the plungers are inclined so that, when the brake is adjusted, the distance of the pivots from the drum centre will vary as little as possible.

It is possible to grind the linings concentric with the drum and true to radius after the shoes have been assembled, by means of an attachment which can be bolted on to the vehicle wheel studs, but the time factor and the dust problem will always militate against this method. Many operators rely on the accuracy of the shoes and the uniformity of the new linings and allow them to bed themselves on the road ; this method can prove surprisingly effective.

Automatic adjustment has, with one exception, been applied only to pivoted shoes, the positive location in the off-position having been considered desirable, if not essential. The exception is significant and automatic adjustment of sliding shoes is likely to be developed in the future.

Sliding Shoes. Because a sliding shoe

can centre itself in the drum, the centre of the wear pattern is not fixed, as with the pivoted shoe, by the motion about its pivot, but depends upon the coefficient of friction between the lining and the drum (Fig. 5). The lining has to be mounted on the shoe in a position which anticipates a certain friction value. If this is exceeded, the lining will wear more at the heel if the shoe is leading or at the toe if it is trailing, and *vice-versa* if the friction is less than expected. In the Girling 2 L.S. design (Fig. 2*b*) the shoes are completely reversible, and the wear pattern is centralized by inclining the face at the shoe tip relative to that at the abutment, but with the parallel abutments of the Lockheed design (Fig. 2*a*), the lining has to be moved round the shoe to the best position. For this reason, the leading and trailing shoes of the latter type are not interchangeable; moreover, the wear pattern is considerably displaced for reverse rotation, but these features do not appear to be detrimental.

In spite of the shift of pressure distribution with change of friction value, the output of sliding shoes is less sensitive to variations of friction than any of the pivoted types, if the theoretical analysis is to be relied on. Errors in the radius of newly relined shoes should have little effect on the output of floating shoes compared with the great sensitivity of the pivoted type in this respect, although the high pressure intensities resulting from localized bedding may have equally harmful effects on the lining material with either type of shoe. In the off-position the shoe is restrained from sliding, as a result of inertia loads, by the friction at the abutment against which the shoe is held by the pull-off spring. In some designs a very strong spring is used in order to provide a large frictional force, but it can be mounted in such a position as to keep its moment about the abutment down to a value which will not make the brakes heavy to operate.

Sliding shoes and their abutments can probably be produced and serviced more economically than any other type, and this alone could account for the trend in this direction, and perhaps for over-emphasis of their functional advantages, material as they are.

Articulated Shoes. The Huck type (Fig. 3*a*) of articulated floating shoes has very similar characteristics to those of a sliding shoe with the abutment face at right angles to the centre-line of the link; the slotted shoe (Fig. 3*b*) belongs in the same category, but the Lockheed tank type (Fig. 3*c*) yields to the same analysis of forces as a pivoted shoe except that concentricity is guaranteed. One of the advantages of the articulated type over the sliding shoe is that by applying a considerable amount of friction at the joint between the shoe and the link, and none at the anchor pin, the shoe can be located, when the brakes are released, exactly as if it were a pivoted shoe. The trend, however, is towards sliding shoes, and while articulated shoes have been used extensively and successfully, they can for the purposes of this classification be regarded as hybrids, the main issue being a comparison between pivoted and sliding types. The indeterminate position of the sliding shoe when the brakes are released has not led to any serious trouble in practice, which gives further justification for the preference for the sliding type of floating shoe, in spite of the advantages of the articulated types referred to above.

Shoe Factors. The "shoe factor" F can be defined as the total peripheral drag at the rubbing surface, as obtained when unit force is applied to the shoe tip, and it is a function of the coefficient of friction μ between the lining and the drum.

$$F = \frac{\text{drum drag}}{\text{shoe-tip load}}$$

The majority of the works referred to in parentheses give detailed methods of arriving at the shoe factors. Using these methods and making certain assumptions which do not change the general form of the equations, it is possible to express the factors of pivoted shoes and of sliding shoes with parallel abutments in the simple formulae shown in Table II, as obtained in sections 1 and 3 of Appendix I and referring to Figs. 5 and 14.

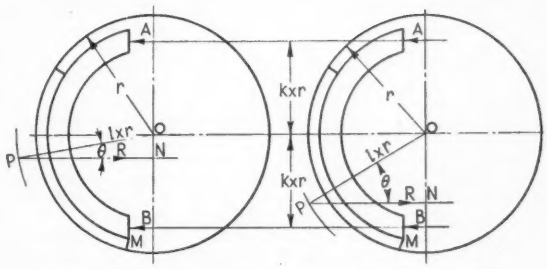


Fig. 5. The Geometry of the Sliding Shoe with Parallel Abutments.

Effect of change of angle of friction on the position of the centre of pressure and point of maximum wear.

The assumptions made in the case of the pivoted shoe are that the line through the centre of pressure and the drum centre is at 90 deg to the line through the pivot and the brake centre, and that the perpendicular distance from the pivot to the line of action of the shoe-tip load is equal to $2k \times r$, as shown in Fig. 14.

In the case of the sliding shoe, it is assumed that the shoe tip and the abutment forces are equidistant from the centre, as shown in Fig. 5, where OA = OB = $k \times r$.

The value of l depends on the included angle of the lining and is likely to lie between 1·1 and 1·17 (Barford 1933). The maximum value of k is limited by the necessity to house the pivot or abutment within the brake drum. It is difficult for it to exceed 0·8, and 0·75 is a typical value. Leading shoes will "sprag" or become self-applying when $\cot\theta$ or $\operatorname{cosec}\theta$ equals l/k, as the case may be. It follows that maximum stability is obtained when the shortest possible lining is used, and when the pivot or abutment is as close as possible to the drum. The maximum output is obtained when the reverse is the case. The question therefore arises as to which is the correct basis on which to compare the two types of shoe, the alternatives being to fix one of the following:—

(1) The value of l and k, say, 1·15 and 0·75.
(2) The value of μ at which the shoe will sprag, say, 0·73.
(3) The shoe factor at a predetermined value of μ, say, $F = 2{\cdot}0$ at $\mu = 0{\cdot}4$.

Three sets of values are laid out on this basis in Table III. The factor at $\mu = 0{\cdot}3$ is also given, and the ratio of this to the factor at $\mu = 0{\cdot}4$ is a measure of the "μ-consciousness" of the shoe within this range. The conclusions which can be drawn from the table are that the sliding shoe tends to have a lower output than the pivoted type and to be more stable, but that it is possible to obtain a very similar performance with either type of shoe, by suitable selection of l and k.

The factors of floating shoes of the Huck type (Fig. 3*a*) and those with sliding abutments inclined to one another (Fig. 2*b*) cannot be expressed as simple formulae but they are easily obtained by graphical methods. They are plotted in Fig. 6 for a typical shoe, of which the shoe-tip and abutment forces act at 21 deg to each other, together with the factors of pivoted shoes,

TABLE I. Bedding of Pivoted Shoe.

	Position of bedding		
	Lining radius greater than drum radius	Lining radius less than drum radius	Lining radius equal to drum radius
Lining equally disposed on shoe ..	Toe and heel	Central	Uniform
Ditto and pivot too close to centre of drum	Toe	Towards toe	Toe
Ditto and pivot too far from centre of drum	Heel	Towards heel	Heel
Lining disposed towards toe ..	Toe	Central	Uniform
Ditto and pivot too close to centre of drum	Toe	Towards toe	Toe
Ditto and pivot too far from centre of drum	Toe, toe and heel, or heel, according to size of error	Towards heel	Heel
Lining disposed towards heel ..	Heel	Central	Uniform
Ditto and pivot too close to centre of drum ..	Heel, heel and toe, or toe, according to size of error	Towards toe	Toe
Ditto and pivot too far from centre of drum	Heel	Towards heel	Heel

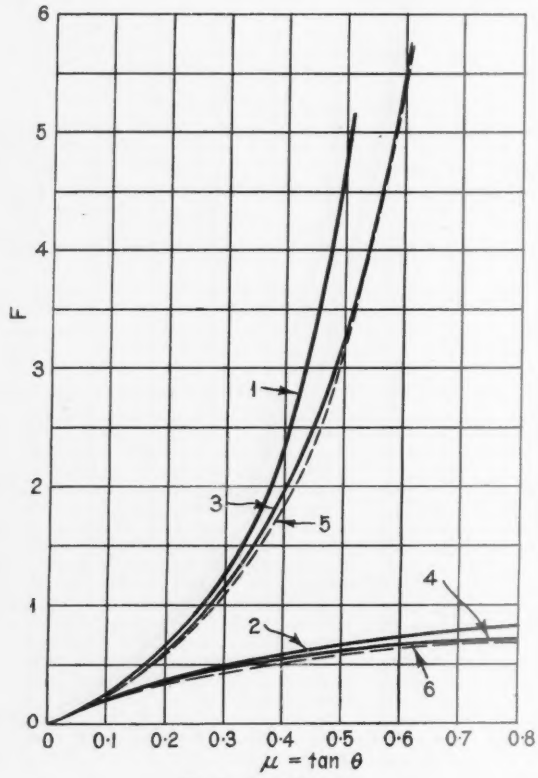


Fig. 6. Shoe Factors.

$F = \frac{\text{drum drag}}{\text{shoe-tip load}}$ $l = 1{\cdot}15$ and $k = 0{\cdot}75$

Pivoted shoes ... 1 leading, 2 trailing
Sliding shoes with parallel abutments ... 3 leading, 4 trailing
Sliding shoes or Huck type with 21 deg. abutments ... 5 leading, 6 trailing

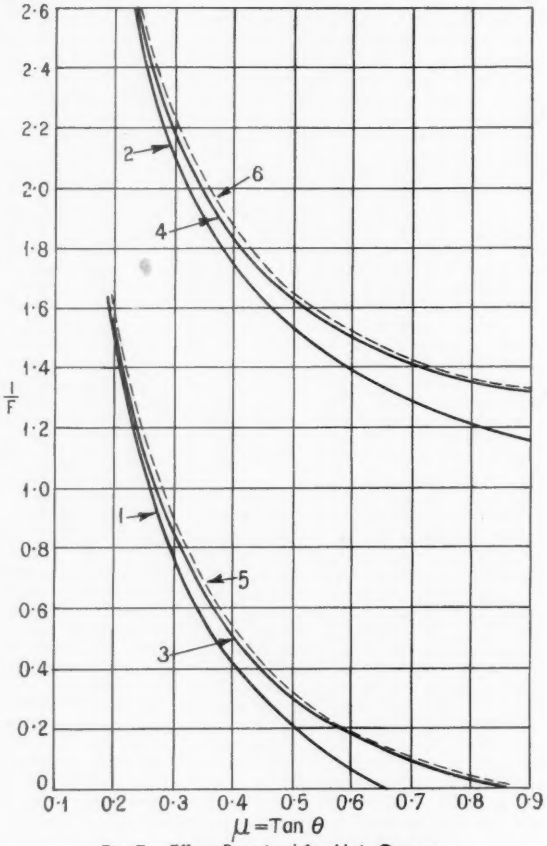


Fig. 7. Effort Required for Unit Output.
(Reciprocals of shoe factors).
$l = 1{\cdot}15$ and $k = 0{\cdot}75$.

Pivoted shoes ... 1 leading, 2 trailing
Sliding shoes with parallel abutments ... 3 leading, 4 trailing
Sliding shoes or Huck type with 21 deg. abutments ... 5 leading, 6 trailing

and sliding shoes with parallel abutments, with $l = 1{\cdot}15$ and $k = 0{\cdot}75$ in each case. The relative inclination of the abutments does not affect the value of μ at which the leading shoe will sprag, which will occur when cosec $\theta = l/k$ for all types of floating shoe. This is because the shoe-tip load becomes virtually non-existent at the spragging point.

In the Lockheed tank type (Fig. 3c), spragging occurs when cot $\theta = l/k$ where k is taken, as it must be for the factor analysis, as the distance OB from the brake-drum centre to the pivot of the carrier. At the spragging point, however, the shoe-tip load disappears and the carrier becomes completely analogous to the link of the Huck shoe, and if k is now measured as OC, the perpendicular distance to the line through the two articulations of the shoe, then cosec $\theta = l/k$ as for the other floating types.

It will be noted that the factor of the trailing shoes is considerably less subject to the μ-value than that of the leading shoes, the equivalent value of $F_{0\cdot4}/F_{0\cdot3}$ being 1·2 (compared with values of 1·7–1·94 in Table III), and it differs but little for the three types of shoe. The marked difference in output between the leading and trailing shoes will also have been noted, and the contrast becomes rapidly greater with increasing friction value.

In Fig. 7 the reciprocals of the factors are plotted against μ for the same three types, so that :—

$$\frac{1}{F} = \frac{\text{shoe-tip load}}{\text{drum drag}}$$

The advantage of this method is that the value of μ for spragging is indicated by an intersection with the base line instead of by an asymptote. The curves are now asymptotic to $\mu = 0$, but it is not necessary to make any reference to them below $\mu = 0{\cdot}2$.

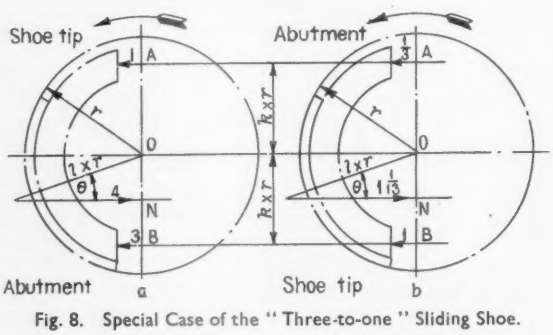


Fig. 8. Special Case of the "Three-to-one" Sliding Shoe.

(a) Leading shoe. (b) Trailing shoe.
$r = 1$; $l = 1{\cdot}12$; $k = 0{\cdot}75$; $\mu = \tan\theta = 0{\cdot}35$.
If shoe-tip load = 1, abutment load = 3 for leading shoe or ⅓ for trailing shoe.
Leading-shoe factor = $3k - k = 1{\cdot}5$.
Trailing-shoe factor = $k - \frac{1}{3}k = 0{\cdot}5$.

Types of Brake. There are four recognized types of internal expanding brakes in common use on road vehicles :—

(1) Leading- and trailing-shoe with floating expander.

(2) Leading- and trailing-shoe brake with fixed cam.

(3) Two-leading-shoe brake.

(4) Self-energizing, or duo-servo brake.

It is possible for any of these types to be equipped with either pivoted or floating shoes, so that from a theoretical standpoint they should be compared on the basis of a common shoe. The great simplicity of the force diagram of the sliding shoe with parallel abutments commends this type as suitable for this comparison. Taking the values $l = 1{\cdot}12$, $k = 0{\cdot}75$, and $\mu = 0{\cdot}35$, the force at one end of the shoe is exactly three times the parallel force at the other end, and the drum drag can be deduced from the carrier plate reactions, being equal to the difference between them multiplied by k (Fig. 8). A twofold comparison will be made on the basis of this common shoe, first an analysis of the factor of each type of brake at $\mu = 0{\cdot}35$, using only simple arithmetic, and secondly the

TABLE II.

	Leading	Trailing
Pivoted shoes	$\frac{2l}{\cot\theta - l/k}$	$\frac{2l}{\cot\theta + l/k}$
Sliding shoes with parallel abutments ..	$\frac{2l}{\operatorname{cosec}\theta - l/k}$	$\frac{2l}{\operatorname{cosec}\theta + l/k}$

where $\tan\theta$ = the coefficient of friction between liner and drum, μ.
l = the distance in terms of drum radius from the centre of the brake drum to the centre of pressure.
k = the distance in terms of drum radius from the centre of the brake drum to the pivot, or, in the case of a floating shoe, the perpendicular distance from the centre to the line of action of the abutment force.

determination of curves for the whole range of values of μ, plotted as the reciprocals of the factors.

Brake Factors. Brake factors have been expressed in various ways. Some quote the sum of the shoe factors, others their mean value; and it is not unusual to find the velocity ratio of the expander included so that the factor is quoted in terms of the pull in the actuating rod. It is therefore essential to find out which method is used when calculating the overall ratio between the driver's effort and the total drum drag. For the purpose of the present comparison the mean of the shoe factors is quoted, so that the brake factor can be defined as :—

$$F = \frac{\text{total drum drag}}{\text{total shoe-tip load}}$$

Leading- and Trailing-Shoe Brake with *Floating Expander.* The simplest, and probably the original example of this type is the hydraulic brake. The equality of loading and freedom of movement of the shoe tips is restricted only by the friction of the pistons and rubber caps in the wheel cylinder. This frictional force and also the frictional forces at the sliding abutments are ignored in the present calculations. The force diagram for this type of brake is shown in Fig. 9*a*, from which it will be clear that the lining on the leading shoe can be expected to wear at least three times as quickly as that on the trailing shoe. This difference can be reduced by increasing the diameter of the piston which operates the trailing shoe, but this will reduce the brake factor because the leading shoe piston will have to be reduced in diameter in order to maintain the original total shoe-tip load. A brake of this type with equal shoe factors at $\mu = 0{\cdot}35$, and therefore equal lining wear as long as this coefficient of friction is maintained, is shown in Fig. 9*e* and curves are shown in Fig. 10 for both straight and stepped cylinders, as well as for the other three types of brake.

In the Girling brake an additional force is acting in the direction of the trailing-shoe tip, equal to the pull in the rod multiplied by the coefficient of friction between the expander housing and the carrier plate on which it is intended to float. With this type the contrast in wear between the leading and trailing

TABLE III. Comparison of Leading Shoes, Pivoted, and Sliding with Parallel Abutments

(The common features in each case are in heavy type)

Type of shoe	l	k	$F_{0\cdot3}$—shoe factor at $\mu = 0{\cdot}3$	$F_{0\cdot4}$—shoe factor at $\mu = 0{\cdot}4$	$\frac{F_{0\cdot4}}{F_{0\cdot3}}$	Value of μ for spragging
Pivoted ..	**1·15**	**0·75**	1·28	2·4	1·88	0·65
Sliding ..	**1·15**	**0·75**	1·18	2·0	1·7	0·855
Pivoted ..	1·1	0·8	1·13	1·95	1·73	**0·73**
Sliding ..	1·15	0·68	1·29	2·5	1·94	**0·73**
Pivoted ..	1·1	0·79	1·13	**2·0**	1·77	0·72
Sliding ..	1·15	0·75	1·18	**2·0**	1·7	0·855

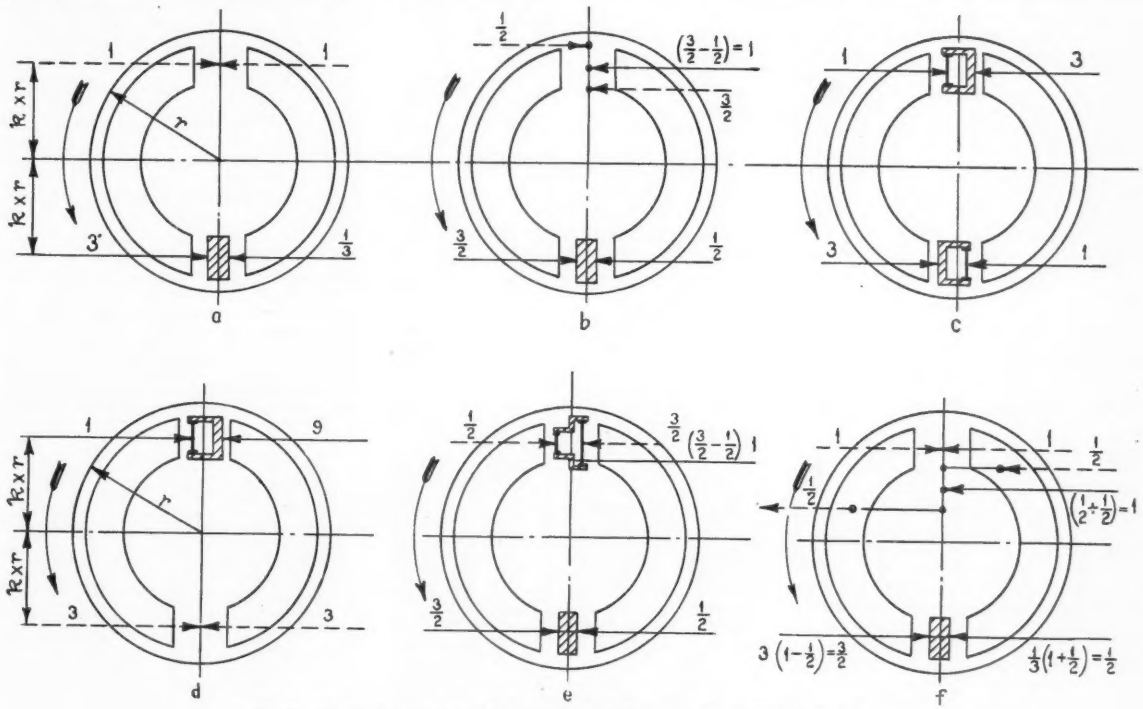


Fig. 9. Force Diagrams for Brakes Using " Three-to-one " Sliding Shoes.

(*a*) Floating expander. $F = (3 - \frac{1}{2})0{\cdot}75/2 = 1{\cdot}0$.
(*b*) Fixed cam. $F = (\frac{3}{2} - \frac{1}{2} + 1)0{\cdot}75/2 = 0{\cdot}75$.
(*c*) Two-leading shoe. $F = (3 - 1 + 3 - 1)0{\cdot}75/2 = 1{\cdot}5$.
(*d*) Duo-servo. $F = (9 - 1)0{\cdot}75/2 = 3{\cdot}0$.
(*e*) Stepped cylinder. $F = (\frac{3}{2} - \frac{1}{2} + 1)0{\cdot}75/2 = 0{\cdot}75$.
(*f*) Hydraulic fixed cam. $F = (\frac{3}{2} - \frac{1}{2} + 1)0{\cdot}75/2 = 0{\cdot}75$.

-------> Reactions on carrier plate. -------> Shoe reactions not acting directly on carrier plate. Drum reactions are not shown.

F = brake factor = $\frac{\text{total carrier plate reaction} \times k}{\text{total shoe-tip load}}$

ROAD VEHICLE BRAKES (*Continued*)

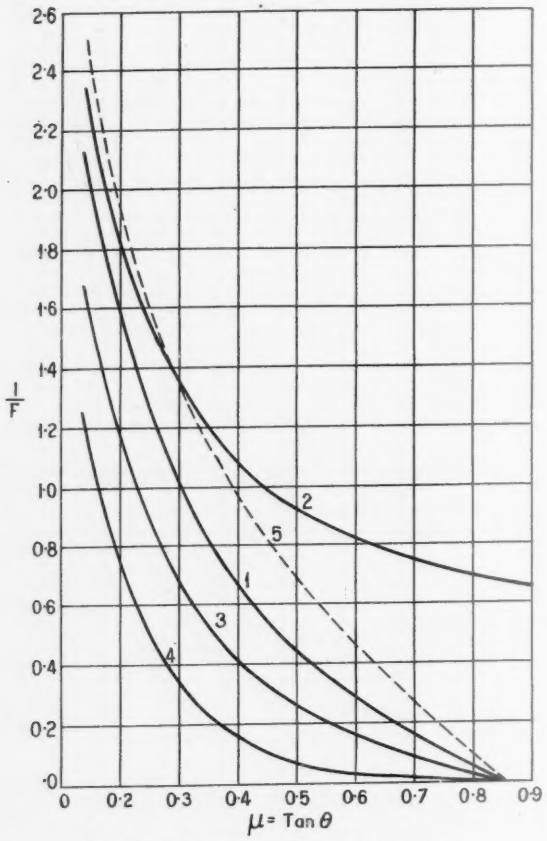


Fig. 10. Effort Required for Unit Output.

(Reciprocal factors of brakes with sliding shoes having parallel abutments.)
$l = 1{\cdot}12$ and $k = 0{\cdot}75$

1. Floating expander.
2. Fixed cam.
3. Two-leading shoe.
4. Duo-servo.
5. Stepped cylinder (3/1).

shoe linings seems to be less marked than with the hydraulic brake, and this frictional force restraining the movement of the expander may be the reason. The effect would also tend to reduce the brake factor.

Other examples of the floating expander type are the Scammell toggle-operated brake and the new Laycock Neal air brake. Pivoted shoes are general, floating shoes having appeared only on some of the hydraulic brakes of this type.

It will be seen that all these brakes will sprag at the same coefficient of friction as would a single leading shoe.

Leading- and Trailing-Shoe Brake with Fixed Cam. By the use of a fixed cam the leading and trailing shoes are forced to move equal distances and therefore have equal lining wear. It necessarily follows that the two shoes must do equal work and therefore have equal factors, which when conjoined give a brake factor of $2l \sin \theta$ with sliding shoes with parallel abutments, or $2l \tan \theta$ with pivoted shoes. The latter is a straight line, since $\tan \theta = \mu$, whilst the former tends towards a maximum value of $2l$ as μ approaches infinity, indicating once again the added stability of sliding shoes, although with this type of brake spragging should never occur whatever shoes are used.

These results are only true as an average during the life of the linings, the varying deflexions at different operating loads being likely to give instantaneous results which are quite unpredictable. It also usually happens that the cam imparts unequal movements, so that its direction of rotation relative to that of the drum is important, but this has been fully dealt with elsewhere (Dawtrey 1930) and the effect is omitted from the present calculations. Equal movements can also be obtained by inter-coupling the shoes by a separate linkage and using a floating expander, and this method makes possible fixed cam characteristics with hydraulic operation (Fig. 9*f*).

The diagram for the type is shown in Fig. 9*b* and the curve appears in Fig. 10. The low output and great stability compared with the identical brake with floating expander will be noted. With an undue rise in friction value it would presumably be possible for the leading shoe to become self-applying and to sprag by leaving the cam behind, unless the two shoes were positively interlinked, with a tension member to hold back the leading shoe, but in practice this seldom happens with a fixed cam brake.

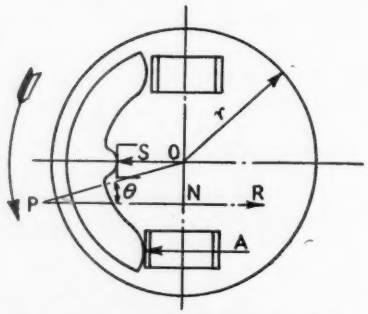


Fig. 12. Suggested Scheme for Hand Brake for Double-acting Two-leading-shoe Hudraulic Brake

$OP = l \times r$; $OA = k \times r$; $ON = l \times r \sin \theta$

$$\text{Factor} = \frac{R}{S} \times \frac{ON}{r} = \frac{OA}{AN} \times \frac{ON}{r} = \frac{k \sin \theta}{k - \sin \theta} = \frac{l}{\operatorname{cosec} \theta - l/k}$$

Two-leading-shoe Brake. This brake has exactly the same characteristics as a single-leading shoe (Fig. 9*c*). The drum drag is doubled but twice as much aggregate shoe tip movement has to be imparted, so that the factor remains unchanged. When separate hydraulic cylinders are used (Fig. 11*a*), the two shoes are fully compensated for both load and movement so that the drum reactions are balanced and produce no off-set load on the wheel bearings, whilst the two linings will run at approximately the same temperature and wear equally, but with the mechanical type using bell crank levers and a strut (Fig. 11*c*), frictional losses will prevent exact equalization.

The hydraulic two-leading-shoe brake can be made double-acting (Fig. 11*b*) if the same factor is required for reverse as for forward rotation; otherwise the brake will be about three times as powerful going forwards as backwards. However, the fashion is to use single-acting brakes on the grounds of economy and to confine them to the front of the vehicle, using leading- and trailing-shoe brakes with floating expanders at the rear. With this arrangement, the overall reverse braking is

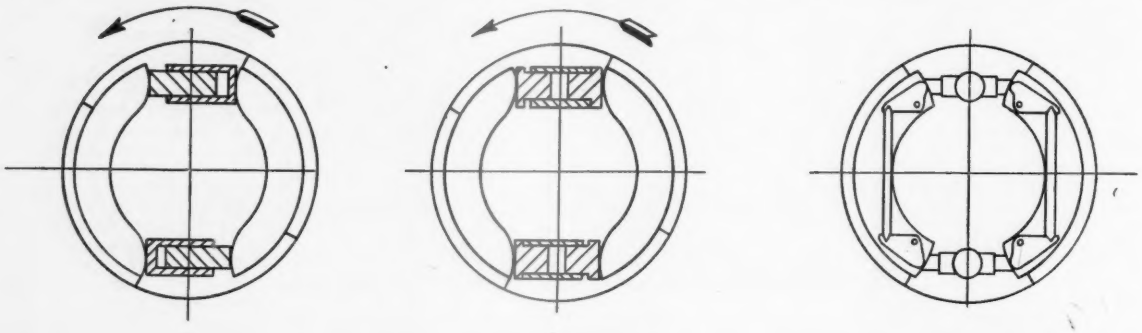


Fig. 11. Two-leading-shoe Brakes.

(*a*) Single-acting hydraulic. (*b*) Double-acting hydraulic. (*c*) Double-acting mechanical.

TABLE IV.

	F	$1/F$
(1) Floating expander	$\frac{2l \operatorname{cosec} \theta}{\operatorname{cosec}^2 \theta - l^2/k^2}$	$\frac{\operatorname{cosec} \theta}{2l} - \frac{l \sin \theta}{2k^2}$
(2) Fixed cam ..	$2l \sin \theta$	$\frac{\operatorname{cosec} \theta}{2l}$
(3) Two-leading shoe ..	$\frac{2l}{\operatorname{cosec} \theta - l/k}$	$\frac{\operatorname{cosec} \theta}{2l} - \frac{1}{2k}$
(4) Duo-servo ..	$\frac{2l \operatorname{cosec} \theta}{(\operatorname{cosec} \theta - l/k)^2}$	$\frac{\operatorname{cosec} \theta}{2l} + \frac{l \sin \theta}{2k^2} - \frac{1}{k}$
(5) Stepped cylinder (3/1) ..	$\frac{2l \operatorname{cosec} \theta - l^2/k}{\operatorname{cosec}^2 \theta - l^2/k^2}$	$\frac{\operatorname{cosec}^2 \theta - l^2/k^2}{2l \operatorname{cosec} \theta - l^2/k}$

approximately three-fifths of normal, so that the pedal load which gives retardation of 12·88ft per sec per sec on the level when travelling forwards will suffice to hold the vehicle from rolling back on a 1 in 4 hill. The overall gain in brake factor on the basis of the " three-to-one " sliding shoe when using two leading shoes at the rear as well as at the front is in the ratio $\frac{300}{(200 + P)}$ where P is the percentage of front braking.

In order to derive the maximum economy from the single-acting type, it has been necessary to develop a method of finishing the blind bores of the cylinders as cheaply as straight-through bores, and the use of die-cast aluminium cylinders has also been introduced.

One of the advantages of the mechanical type of two-leading-shoe brake using the bell cranks and strut is that it lends itself for dual operation by hand or foot, so that it can be employed at the rear as well as at the front; moreover, double action can be used if necessary without upsetting the hand-brake linkage. It is, however, difficult to apply the hand linkage to both shoes of the double-acting, hydraulic type because it requires a separate mechanical expander at each end of the shoes, and a fully compensated linkage between them in order that hand and foot operations shall not conflict. Such a design was produced by Alfred Teves of Frankfurt, but nothing of the kind has gone into production in this country. A simple alternative would be to apply the hand brake load midway along one shoe only in each rear brake (Fig. 12). This would give a factor for forward or reverse rotation of $\frac{l}{\operatorname{cosec} \theta - l/k}$, which is half that obtained when the load is applied to the tip of a leading shoe. With such an arrangement the expander would only have to impart half the movement which would be required at the shoe tip, so that the normal hand-brake velocity ratio could be doubled, and the resulting brake would have the same overall factor and charactieristics as a double-acting, two-leading-shoe brake. The wear on the single lining would be accelerated according to the use made of the hand brake, and the brake assembly would have to be capable of dealing with the off-set load.

The practical difficulties of this scheme would be to provide the necessary loads and movements within the space available, together with a means of adjustment for lining wear, while keeping the parts simple and therefore inexpensive.

The two-leading-shoe brake with two internal cylinders would appear to be the ideal type because of the perfectly balanced duty of the two shoes, and a hand-brake linkage enabling such a brake to be used on the rear wheels is overdue. If the temperature inside the brake drum is above the boiling point of the hydraulic fluid, mineral oil and synthetic rubber cups can be used, although a far more desirable plan, which will be discussed later, is to provide air cooling inside the brake drum.

Amongst other examples of two-leading-shoe brakes are those produced for heavy vehicle axles by Kirkstall Forge, Ltd. Sliding shoes are used with a single mechanical expander, and a beam which applies the load midway along the " trailing " shoe on the same principle as that proposed for the hand brake with the hydraulic type. The two-leading-shoe brake has also been favoured by Fodens, Ltd.

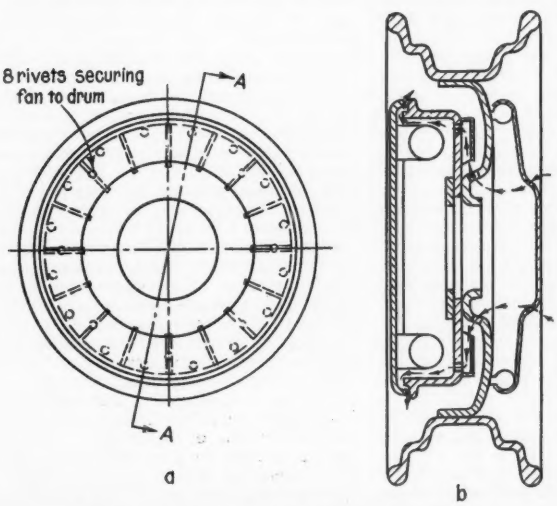


Fig. 13. Proposed Cooling Fan for Brake-Path.
(*a*) End view of drum and fan. (*b*) Section AA.

Self-energizing or Duo-servo Brake. This brake (Fig. 9*d*) has the highest factor of any of the internal-expanding types and is used to give the lightest possible pedal-load without having to resort to power-assistance. Operated by a direct-acting, internal hydraulic cylinder, it is fitted to many American cars, on which it appears to be quite satisfactory. Mechanically operated, it has a less enviable reputation, because it is a super-sensitive brake and

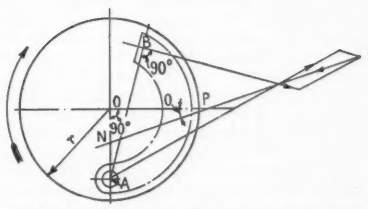


Fig. 14. Geometry of a Pivoted Leading Shoe.
$OA = k \times r$; $AB = 2k \times r$; $OP = l \times r$.

frictional losses in the linkage and expander mechanism cannot be tolerated; moreover linings with a stable friction value are essential.

As with the leading- and trailing-shoe brake with floating expander, the secondary or energized shoe can be expected to wear at least three times as quickly as the primary or energizing shoe. To reduce this effect, and to stabilize the brake, special low-coefficient linings are sometimes fitted to the secondary shoe, a course which is permitted by reason of the very high factor of this type of brake. It will be noted that, in spite of the high factor and " μ-consciousness " of the duo-servo brake, the spragging point is exactly the same as for the other types with free shoes. In determining the brake factor, the total shoe-tip load is taken as 2, whereas unit load is actually applied to the tip of the primary shoe only. In order to take up the clearance of the secondary shoe, the tip of the primary shoe has to move twice as far as would otherwise be the case, and in the static state, with the drum free to rotate, the unit load is carried round to the tip of the secondary shoe. Thus it follows that the total shoe-tip load must be given a value of 2, for the factor to be directly comparable with the other types.

The duo-servo brake is often arranged so that the primary shoe floats while the secondary shoe is pivoted, this being true for reverse as well as for forward rotation. There would, however, appear to be a strong case for making both shoes float, in order to increase the stability and raise the spragging point, and the theoretical diagram shown in Fig. 9*d* and the curve shown in Fig. 10 are for a brake of this type.

It has always been a problem to centralize duo-servo shoes in the off-position, and various devices have been introduced, including the use of independent pull-off springs of different poundages for each shoe.

Comparison of Brake Factors and Characteristics. The factors of the types of brakes described, when compared on the basis of the sliding shoe with parallel abutments, can be expressed by the formulae shown in Table IV as determined in Appendix I :—

It is from these equations that the curves in Fig. 10 are plotted, and the comparative outputs and degree of μ-consciousness as well as the value of μ for spragging can readily be obtained. In practice there are many influences which cause the brake performance to depart from the calculated curve and the features which appear to be most desirable are :—

(*a*) A reduction to the smallest possible value of the friction in the entire linkage between the pedal and the shoes—this demands freedom from geometric quarrel due to relative movement of the chassis components, and that is obtained more easily by using hydraulic or pneumatic systems than with mechanical linkages.

ROAD VEHICLE BRAKES (Continued)

(*b*) Full compensation between the movement of the two shoes in each brake to enable them to follow displacements of the drum with minimum effect on the applied loads and reactions. This is possible with all types except the fixed cam brake, but is difficult to achieve except by hydraulic or pneumatic operation applied direct to the shoe tips.

(*c*) Shoes which can centre themselves in the drums under all working conditions, that is to say, floating shoes of various types.

(*d*) Uniform distribution of work over the linings. This demands equal shoe factors, which are obtained more consistently with the two-leading-shoe brake than with the fixed cam or stepped cylinder. It also postulates a uniform pressure-distribution over the lining surface but this is only possible with flexible shoes and not easy to achieve satisfactorily, particularly with long linings. It has often been recommended that the included angle of the liner should not be greater than 90 deg, but the advice is seldom followed, 110 deg being a popular figure.

(*e*) Adequate cooling of the brake-path. Short linings also help in this respect because a considerable amount of heat is dissipated from the brake-path by radiation from the uncovered portions between the shoes. The biggest improvement in brakes which can be looked for in the future is the development of air cooling of the brake-path and this will be discussed later.

Such features as the above can have at least as much effect on the stability of the brake as the characteristics of the factor curve, and the two-leading-shoe brake, with a separate internal cylinder for each floating shoe, seems likely to give the most consistent results of any type. If a higher output is needed without using power-assistance, the duo-servo brake can be made considerably less temperamental than this type has been on English vehicles, as the Americans have demonstrated.

Outstanding Braking Problems. The incomplete knowledge regarding the nature of friction and the characteristics of the organic materials used in the manufacture of brake linings combine to prevent the exact evaluation of friction braking and its associated phenomena. There are also mechanical problems which demand further development. The outstanding subjects which single themselves out for special mention as requiring further research and development are :—

(1) Brake cooling.
(2) Lining and drum wear.
(3) Brake squeal.
(4) Temperature fade of linings.
(5) Automatic adjustment.
(6) Escape of lubricant from hubs on to brake linings.

Brake Cooling. All the work which is turned into heat eventually appears as a rise in the temperature of the atmosphere. The higher the temperature of the brake drum, the better will be the heat transfer to the surrounding air, but unfortunately neither the brake linings nor the tyres of the vehicle will operate above certain critical temperatures, and there are also limits to the temperature at which the brake-track of the iron drum will operate satisfactorily. The usual way to keep the temperature down is to put plenty of mass in the drums and to use them as reservoirs from which the heat can be dissipated during the periods between the operation of the brakes, but the comparatively low thermal conductivity of the iron results in a steep temperature-gradient and very high surface temperatures unless the rate of energy absorption is strictly limited.

It follows that an induced draught, direct to the brake-path where it is exposed between the shoe, is the only effective way of cooling brakes with a high rate of energy absorption. Attempts to direct jets of air on to the brake-path have not been very successful, the reason probably being that the actual weight of air emerging from the jet has been comparatively small.

What is required is a fan which will pump a really substantial weight of air through the inside of the brake. The only feasible way of doing this appears to be to form a centrifugal impeller on the drum backplate, drawing air through the centre of the road wheel and discharging it through orifices near the outer periphery of the brake carrier-plate, as in Fig. 13. The effectiveness of such a scheme is seriously limited by the low rotational speed of the road wheel, particularly on 'buses and trucks, where improved cooling is most needed, but it appears to be the only feasible way by which air can be circulated within the brake. Forward facing air scoops are used on racing cars, but are ineffective on slow-moving vehicles with shrouded wheels.

Lining and Drum Wear. As far as is known, the rate of wear of all rubbing solids increases with rise of temperature, unless, of course, the coefficient of friction falls so much that the materials save themselves, preventing any further temperature rise. Brake linings are no exception. The best possible cooling is therefore imperative, but apart from this there is no doubt that wide variations in rate of wear, and in the rate of increase of wear with rise of temperature, exist between different lining materials working at equal absorptions. Moulded materials offer greater scope for improvement in this respect than do the woven types, and much is being learned, but so wide is the field of materials which can be incorporated in brake linings that it is likely to be a long time before their characteristics can be exactly defined and controlled, as is the case, for example, with steel. Whatever the nature of the linings, a highly polished drum surface is necessary if wear is to be reduced to a minimum. In increasing the life of the lining, care has to be taken that the wear has not been transferred to the drum.

Brake Squeal. From various theories and tests and from palliatives which have been tried, the following conclusions may be reached :—

(1) Sliding friction can occur with or without "stick-slip." The stick-slip results from the static friction being greater than the kinetic friction, and it is accentuated by the presence of thin films of certain liquids. The phenomenon occurs more readily with smooth polished surfaces than with rough ones. In cases where stick-slip has been absent, as when rubbing like metals together, there has been a tearing of the surface and general disintegration. It is not known whether at the other end of the scale friction can exist entirely as stick-slip without any wear at all, but there is a wide field here open for research. When stick-slip occurs, relative rest must be produced during each cycle, and a vibration of the deflected parts is induced.

(2) Squeal occurs when the stick-slip excites some natural frequency in the brake assembly which falls within the audible range. Experiments with a brake drum which was progressively reduced in mass showed no appreciable change in the squeal frequency (Motor Industry Research Association 1939), but changing the mass and stiffness of the lining holder or shoe have produced changes in the pitch of the note. It is suggested that the shoe assembly should be regarded as a tuning fork and the drum as a sound box, and this theory requires verification.

(3) One individual brake may squeal intermittently. It is reasonable to conclude that the only thing in the system which would be likely to undergo a change which could alter these characteristics is the state of the rubbing surfaces. A study of squeal is therefore likely to resolve itself into a study of the nature of the rubbing surfaces under different conditions of service.

(4) A high intensity of work on part or all of the lining appears to be conducive to producing a squeaky surface. The removal of shiny high spots from newly fitted linings which have squeaked initially has frequently effected a permanent cure. When the whole surface has become polished after a period of service, roughing it up has only effected a temporary cure.

(5) Squeal is usually associated with a high coefficient of friction and freedom from squeal may demand a limiting friction value.

(6) The most generally successful palliative is the "anti-squeak band" which consists of asbestos tape or other non-resonant material wrapped round the brake drum and secured by means of a strip-steel clamp.

(7) Successful results have been obtained by cutting away the web of the shoe to introduce local flexibility, the position and dimensions of the slot thus formed being critical. This will alter the pressure distribution over the lining and possibly reduce the maximum local intensity of work by means of a more uniform distribution. It will also effect the natural frequencies in the shoe.

A scientific approach to the problem of squeal resolves itself initially into two studies, one of the nature of the rubbing surfaces which may call for an electron microscope, and the other of the exact nature of the vibrations, whose small amplitudes and high frequencies will require something very special in the way of cathode-ray oscillograph technique.

Temperature Fade of Linings. A coefficient of friction which rises with increase of temperature is unstable and there is an inevitable build up of temperature, tending towards hot spots, which will be accentuated by thermal expansion and become self-destructive.

On the other hand, fade is a stable condition which automatically limits the peak temperature. Its exact nature is not known, but since it becomes less marked with repeated heating of the linings, it seems likely that something which can act as a lubricant is formed, and with the phenomenon known as delayed, or "deadly" fade, exudation can sometimes be observed. A small amount of progressive fade with rapid recovery can be tolerated and may even be desirable, but the temperatue of its onset should be above the normal working maximum. Delayed fade is quite intolerable ; a truck may descend a long grade without difficulty and then fail to stop at a cross-road half-a-mile beyond the foot of the hill.

Automatic Adjustment. Automatic adjustment, to be fully effective, must compensate for wear of the linings, but at the same time allow the shoes to follow changes in diameter of the brake drum due to thermal expansion as well as mechanical deflexion ; moreover the adjusters must not be deranged by the dirt which is

inevitably present in the brakes. Further development in this field is clearly needed, and it is unlikely that manual adjustment will survive.

Escape of Lubricant from Hubs on to Brakes. Linings soaked in oil or grease are a familiar sight in any brake service station. Over-lubrication of hubs is sometimes the reason, but the design should render this impossible. The cause is often that heat from the brakes melts the grease, but here again it should not be impossible to design a bearing which could retain its lubricant in a liquid form. The evidence would indicate that here is a problem requiring further attention.

Brake Drum Developments. Special attention should be drawn to bi-metal brake drums in which a coarse-grained iron brake-path is centrifugally cast into a pressed steel shell.

Notes on Brake Testing. Testing brakes on vehicles is a futile pursuit unless the tester is perfectly clear in his mind as to what he is trying to find out, and is also fully aware of the likely limitations of his task. Testing divides itself into studying the normal performance of the system and exploring the departures from this performance which can occur, with their causes.

Whether roads are rough or smooth, wet or dry, the shortest possible stopping distance is most likely to be obtained by locking all the wheels of the vehicle. This is because of the practical impossibility of keeping all four wheels only just off the point of locking throughout the retardation, for the driver cannot anticipate the possible changes in road adhesion or variation in lining friction value, nor can he benefit from the rise in road adhesion with fall in speed which has been shown to occur when the wheels are locked (Gough 1949). In addition to this, variations in the brake distribution are quite outside the driver's control; moreover, the front to rear braking ratio can only be correct for one value of road adhesion.

On certain servo systems, steps are taken to prevent excessive braking on the front or on all of the brakes, and on some heavy vehicles without power-assistance the pedal load required to lock the wheels is greater than the driver's strength, but in general, even if the brakes at one end of the vehicle lock before those at the other, it should be possible to lock all of the wheels by pushing hard enough on the pedal. The simplest form of brake test is, therefore, to find out what happens under maximum braking on both rough and slippery surfaces and, if all the wheels cannot be locked, to find out why.

Brake meters of the Tapley or bubble type can be used to obtain an approximate value of the deceleration, or an accurate analysis of the retardation can be obtained by means of a fifth wheel recording distance on a time base, but in either case much of the value is lost unless a simultaneous reading is taken of the effort applied by the driver to the pedal. This can be done with reasonable accuracy by means of a "pressometer" which consists of a pressure gauge connected to a hydraulic cylinder and piston of 1 sq in cross-sectional area, interposed between the driver's foot and the pedal, thus giving a direct reading of the load in pounds.

Curves should be plotted of retardation against pedal effort, and if a Tapley meter is used, an average for each stop obtained visually will be found to be more informative than using the trip mechanism which records only the instantaneous maximum. It is best to put the readings straight on to graph paper, because discrepancies can then be seen and checked, while deterioration of the brakes during the test will be apparent and readings can be repeated as required. The curve should approximate to a straight line and it should continue as such until the limiting adhesion between the tyres and the road is reached. Its slope will be proportional to the overall brake factor and if, on repeating the test, it becomes progressively less steep, then a fall in the friction of the linings can be suspected, but it is only the change in the brake factor which can be measured.

If the curve flattens out more or less abruptly before any locking of the wheels occurs and yet can be repeated below this value, it will be clear that some mechanical limitation in the linkage or the brake expanders has been reached and must be investigated. Brakes requiring adjustment will give this effect.

Retardation curves of this type can be taken at intervals during the life of the brake linings and in this way a fairly complete history of the brake performance can be obtained. There is, however, always a risk of introducing artificial conditions by overheating the linings beyond their worst service conditions while taking the curves, and this should be guarded against and the test slowed down or discontinued if the slope of the curve falls unduly.

In this connexion temperature measurements ought to be taken, but to be of any real value simultaneous readings of all four (or more) brakes are required, because it is common for one to overheat before the others. Even when this is done it is difficult to distinguish between equal deterioration of all four brakes or complete fade of one or two brakes only, and it is possible that the hottest brake will be the one that does not fade. An attempt is, in fact, being made to carry out four separate brake tests simultaneously, and to analyse them individually, a task which would be difficult enough in the laboratory, not to mention a vehicle travelling along the highway. If the temperatures of all four brakes are recorded, it may be possible to recognize the fade of an individual brake when its temperature ceases to rise, or even falls, while that of the other brakes continues to increase. A fixed cycle of operations on the same test course should be established and at each stage the deceleration will be plotted against pedal effort, together with the brake temperatures, if they have been obtained.

Throughout their working life, the brake linings and drums are undergoing repeated surface treatment under a wide range of rubbing speeds, unit pressures, and temperatures, which produces ever-changing surface finishes and conditions, and it must be remembered that it is the behaviour of the type of surfaces prevailing at the moment which is being tested.

It is the study of surface phenomena and the search for better cooling which form the fundamental basis for progress in the science of braking.

Appendix

DETERMINATION OF FORMULAE FOR SHOE FACTORS AND BRAKE FACTORS

(1) Factors for sliding shoes with parallel abutments.

In Fig. 5, drum drag $= R \times \frac{ON}{r}$.

Shoe-tip load $= L$ for leading or M for trailing shoe.

Therefore $F = \frac{R \times ON}{L \times r}$ or $\frac{R \times ON}{M \times r}$

$= \frac{AB \times ON}{BN \times r}$ or $\frac{AB \times ON}{AN \times r}$

$= \frac{2kl \sin \theta}{k - l \sin \theta}$ or $\frac{2kl \sin \theta}{k + l \sin \theta}$

$= \frac{2l}{\text{cosec } \theta - l/k}$ or $\frac{2l}{\text{cosec}\theta + l/k}$

(2) Factors for brakes using sliding shoes with parallel abutments. The brake factor is the mean of the shoe factors. The brake layouts are shown in Fig. 9.

(*a*) Floating expander :—

$$F = \tfrac{1}{2}\left(\frac{2l}{\text{cosec } \theta + l/k} + \frac{2l}{\text{cosec}\theta + l/k}\right)$$

$$= \frac{2l \text{ cosec } \theta}{\text{cosec}^2\theta - l^2/k^2}.$$

(*b*) Fixed cam :—

Let x = leading shoe-tip load,
and $(2-x)$ = trailing-shoe-tip load;
then $(2-x)$ = leading-shoe-abutment load,
and x = trailing-shoe-abutment load.

In Fig. 9*b*,

$$F = 2\{(2-x) - x\}k/2$$

In Fig. 5,

$$\frac{(2-x)}{x} = \frac{M}{L} = \frac{AN}{BN} = \frac{k + l \sin \theta}{k - l \sin \theta}$$

$$\text{therefore } x = \frac{k - l \sin \theta}{k}$$

and $F = 2l \sin \theta$.

(*c*) Two-leading shoe :—

As for a single-leading shoe,

$$F = \frac{2l}{\text{cosec } \theta - l/k}$$

(*d*) Duo-servo :—

$$\text{Primary-shoe factor} = \frac{2l}{\text{cosec}\theta - l/k}$$

Secondary-shoe-tip load = primary shoe-abutment load

$$= \frac{k + l \sin \theta}{k - l \sin \theta}$$

$$\text{Therefore } F = \tfrac{1}{2}\frac{2l}{(\text{cosec}\theta - l/k)} \times \left(1 + \frac{k + l \sin \theta}{k - l \sin \theta}\right)$$

$$= \frac{2l \text{ cosec } \theta}{(\text{cosec } \theta - l/k)^2}$$

(*e*) Stepped cylinder, ratio $s/1$:—

Let leading-shoe-tip load be x. Then trailing-shoe-tip load is $(s \times x)$ and $x + (s \times x) = 2$.

$$\text{Therefore } x = \frac{2}{(s+1)}$$

$$\text{and } (s \times x) = \frac{2s}{(s+1)};$$

$$F = \tfrac{1}{2}\left\{\frac{2}{(s+1)} \times \frac{2l}{(\text{cosec}\theta - l/k)} + \frac{2s}{(s+1)} \times \frac{2l}{(\text{cosec}\theta + l/k)}\right\}$$

$$= \frac{2l}{(s+1)}\left\{\frac{(s+1)\text{ cosec}\theta - (s-1)l/k}{\text{cosec}^2\theta - l^2/k^2}\right\}.$$

(3) Factors for pivoted shoes with centre pressure at 90 deg. from the line joining pivot and drum centre, and the perpendicular distance from the pivot to the line of action of the shoe-tip load twice the distance from the pivot to the centre.

In Fig. 14, which shows a leading shoe, from Acres,

$$F = \frac{AB}{r} \times \frac{ON}{AN}$$

$$\frac{ON}{OP} = \tan \theta = \mu; \quad OP = (l \times r);$$

$$AB = 2OA = 2kr.$$

$$\text{Therefore } F = \frac{2kr}{r} \times \frac{lr \tan \theta}{(kr - lr \tan \theta)}$$

$$= \frac{2l}{\cot \theta - l/k}$$

Similarly, for a trailing shoe,

$$F = \frac{2l}{\cot \theta + l/k}$$

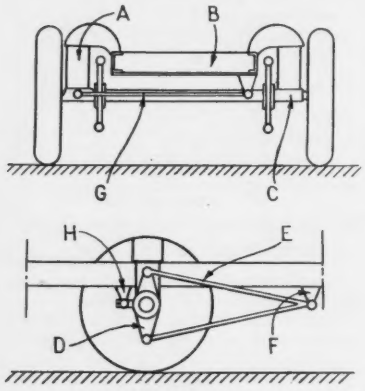
CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Rigid Axle Suspension

THE use of helical springs, torsion rods or pneumatic devices instead of the conventional laminated springs for the suspension of road vehicles is usually associated with independently sprung wheels. Such systems are best suited to vehicles having a centre of mass sufficiently low for the road performance not to be affected by rolling motion. For commercial load- and passenger-carrying vehicles it is proposed to use the lighter and more resilient elastic systems in connection with a rigid axle.

Resilient members A, which may be



No. 610518.

helical springs, rubber blocks, pneumatic or hydro-pneumatic devices are mounted between the chassis or body B and a rigid axle beam C to receive and absorb vertical loads. Rigidly attached to each end of the axle is a lever arm D to which is articulated two rods E having a common articulation point on a frame bracket F. These rods, for which may be substituted a single forked member or a single flexible bar rigidly attached to the axle, prevent horizontal movement of the axle in relation to the chassis and transmit thrust and braking reactions. In the transverse direction stability is ensured by rod G articulated to one end of the axle and to frame bracket H. *Patent No.* 610518. *Electrotube-Solesmes. (France.)*

Rubber Spring Starter

OPERATING on the basis of elastic energy, a rubber spring is wound up by a hand crank, a cable and ratchet or a small electric motor and then released to start an engine. The spring is composed of a number of moulded rubber rings A connected in series and attached at one end to a winding mechanism and the other end to a controlled starter coupling.

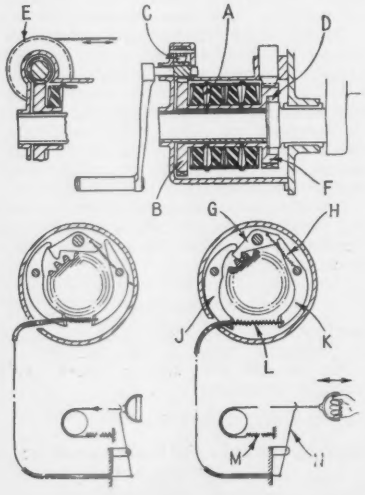
In one example, the winding device consists of a gear wheel B driven by the ratchet-controlled pinion C. The coupling to the engine may be a unidirectional clutch D with wedging rollers. As an alternative method of winding a worm and wheel may be used and operated remotely by means of a cable around a ratchet-controlled drum E. The operating cable, in this instance, may also be used to actuate the latching and release mechanism. Latching member F has a toothed periphery and is engaged by a latch G normally held out of engagement by spring H. Engagement is effected by lever J depressing the latch which is then held by locking lever K. These two members are under the constraint of compression spring L but when the mechanism is at rest this spring is overcome by tension spring M which returns the winding cable.

To effect a start the winding cable is drawn to release lever N, connected by Bowden cable to levers J and K, and thus allow spring M to engage and lock the latch. A series of pulls on the cable revolves the drum a sufficient number of times to stress the spring and, on release of the cable, lever N is engaged to trip the latch and permit the stored energy of the spring to be applied to the engine shaft.

The specification also describes a method in which the spring is wound by an electric motor through a worm gear and the torque reaction is used to close automatically a pair of contacts and effect the energization of a solenoid controlling the latching mechanism. A further refinement of this system is that the position of one of the contacts may be varied by means of a thermal control responsive to the temperature of the oil in the engine sump to ensure that the unlatching is effected only when the spring has been stressed to a value appropriate to prevailing conditions.—*Patent No.* 609088. *S. A. André Citroën (France).*

Variable Pressure Fuel Injection Pump

WITH a view to obviating or minimizing the sudden pressure rise in the combustion chamber consequent upon ignition lag, this injection pump delivers fuel uninterruptedly at a substantially



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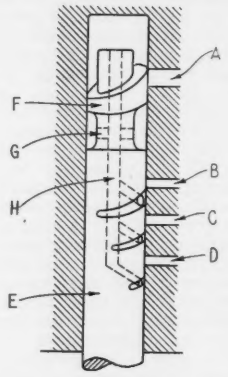
constant rate but at a pressure that is reduced in one or more stages during the injection period. A relatively high pressure at the commencement of injection secures improved atomization and consequent reduction of lag. Subsequent reduction in fuel pressure leads to coarser atomization and steady burning of the remainder of the charge. The lower pressure values are secured by increasing the effective area of the inlet to the combustion chamber and may be brought about simply by having two or more spray passages each controlled by the reciprocating movement of the pump plunger.

The example shows a pump cylinder having an inlet passage A and three de-



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livery passages B, C and D. Pump plunger E has reduced-diameter portions above and below the inlet scroll F. The lower portion is formed by an annular groove which communicates through radial bores G and central bore H with the inlet space above the plunger. Drillings also connect scroll grooves co-operating with delivery passages B, C and D to the central bore.

Fuel is admitted to the cylinder above the plunger by way of inlet port A and as the plunger rises to close the port compression commences. Immediately following, passage B begins to be uncovered by its respective scroll groove and delivery commences at high pressure. Continued upward movement successively uncovers passages C and D so that fuel is eventually delivered through three spray orifices. Injection is terminated when the lower edge of the inlet scroll uncovers port A and relieves the pressure.

The scroll and scroll grooves are appropriately shaped so that partial rotation of the plunger will vary the timing of commencement and termination of injection and consequently the volume of fuel delivered. It will be seen that a plunger of this type renders both the earlier and later portions of the delivery stroke ineffective and the fuel is delivered only in the intermediate part of the stroke. This feature, in conjunction with the successive lowering of pressure, avoids the high peak pressures commonly developed and results in smoother running of the engine. *Patent No.* 610095. *K. Rasmussen. (Denmark.)*